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The design of a four square gear/lubricant research instrument

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THE DESIGN OF A FOUR SQUARE GEAR/LUBRICANT RESEARCH INSTRUMENT

HARRY JAMES HANSEN

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THE DESIGN OF A FOUR SQUARE

GEAR/LUBRICANT RESEARCH INSTRUMENT

by

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Lieutenant, United States Navy
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ABSTRACT

The gear or lubricating oil test instrument described incorporates the four square principle in its design. This basic configuration is improved over like devices in that loading may be varied while the machine is in operation and rotating seals have not been used. The elimination of the need for these seals allows prolonged operation at high speeds. The machine will be capable of operation at 10,000 RPM however, the first phase of operation will call for a speed of 1800 RPM. The device has two separate and complete thermostatically controlled lubrication systems; this gives the option of testing lubricants in the main, as well as the test section. The test section is provided with an observation port which is easily removed to change or service the test gears. The rig as a whole was designed for reliability of operation and ease of maintenance.

ACKNOWLEDGEMENTS

The author wishes to express his sincere appreciation to Professor Ernest K. Gatcombe for the many valuable suggestions he made as consultant for design and for his constant encouragement during the entire project. The instrument was built in the machine shops of the Naval Postgraduate School and special mention is due Mr. Henry F. Perry for the care and skill shown in the fabrication and assembly of the machine.

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1. Introduction.

Man's desires and ambitions have always been far in advance of the materials to fulfill them. We are constantly striving to develop new products and techniques and to improve upon old ones. New vistas of exploration put increased demands on the machines we use. These demands have been felt in the lubrication field and much research is being conducted to combat man's old enemy, wear. The aircraft industry is particularly anxious to learn about the effects of lubricants on gear wear; the effects of viscosity have been studied but the published data is sparse in the range of speeds and loads encountered in turbine driven aircraft. There is a definite need to know if surface treatments to the gears or new compounds may permit higher loads to be carried. All these things should be measured by some kind of numbers so that useful comparisons may be made. /12/

In any research endeavor there are instruments for evaluating the products developed. The design described in this thesis will provide a reliable and relatively easy to operate instrument that will be useful in the study of gear wear and lubrication effects. During the first phase of operation the machine will be run at 1800 RPM to evaluate its performance at loadings up to and including the initial design load of 2650 pounds per inch of gear face width. The load will be varied over the entire range while the machine is in operation to calibrate and check the characteristics of the loading system.

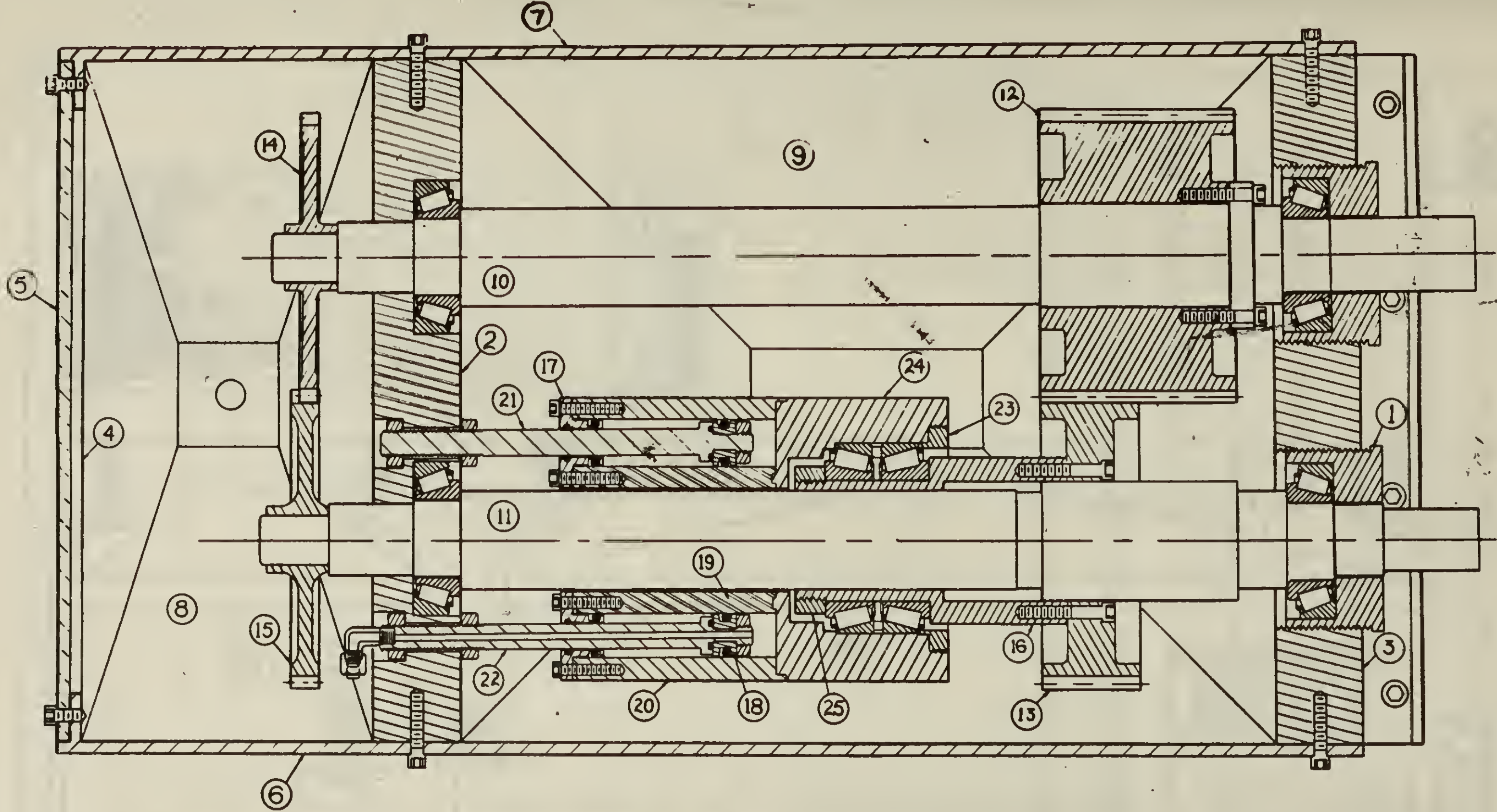
It is felt that the design of this particular machine is an improvement over other similar devices using hydraulic loading in that there are

no rotating seals. This feature is particularly advantageous for prolonged high speed operation.

A plan view, C403, and side view, C402, of the main assembly are shown to acquaint the reader with the instrument before considering the various sections of the thesis. The individual sections are broken down into the design of the main and test gear shafts, hydraulic assembly, lubrication systems, control and hydraulic system, and supporting structure respectively.

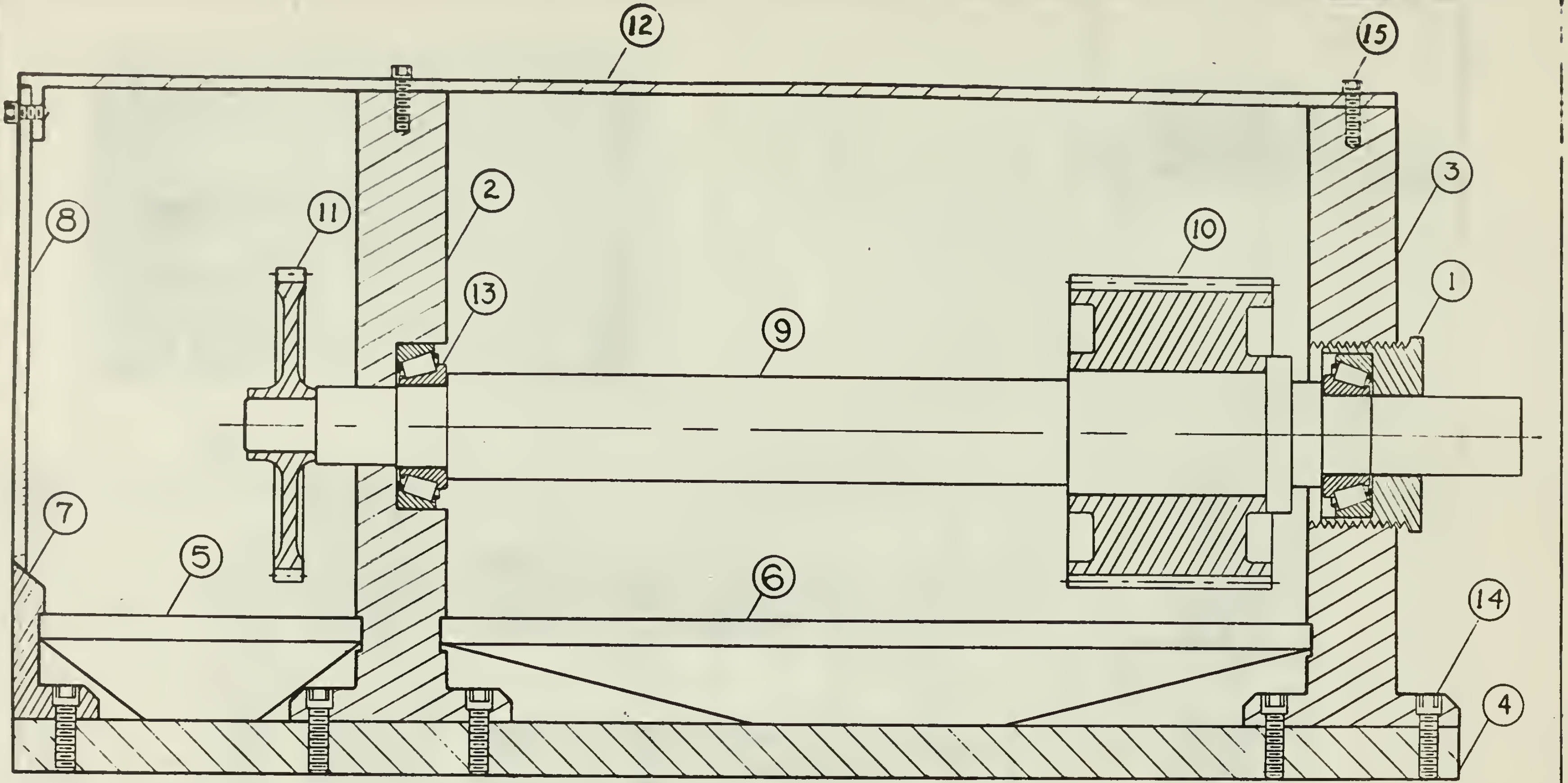
In studying the various drawings in each section, the reader will note that a scale is mentioned on several. In all cases the drawings in the material presented are photo reductions of the plans used to fabricate the various parts. The scale should not be construed to apply to the reproductions appearing in this thesis.

The notation used in the formulas presented is, as far as possible, in accordance with the notation convention established by the American Society of Mechanical Engineers and the American Gear Manufacturers Association. In order to eliminate any confusion in this regard, all symbols are defined in the section in which they are used.



	PART	NO. REQ.	DWG. NO.
1	BEARING ADJUSTER	2	B201
2	TEST END BEARING HANGER	1	B202
3	DRIVE END BEARING HANGER	1	B203
4	TEST SECTION END	1	B205
5	VIEW PORT	1	B208
6	SIDE PANEL HYDRAULIC SIDE	1	B206
7	SIDE PANEL	1	B207
8	TEST SECTION OIL PAN	1	L301
9	DRIVE SECTION OIL PAN	1	L302
10	SHAFT	1	S501
11	DRIVE SHAFT	1	S502
12	DRIVE GEAR	1	G701
13	HELICAL PINION	1	G702
14	TEST GEAR	1	G704
15	TEST SECTION DRIVE GEAR	1	G703
16	RAM	1	H101
17	END PLATE	2	H102
18	PISTON	1	H103
19	INNER WALL	1	H104
20	OUTER WALL	1	H105
21	PISTON ROD	2	H106
22	PISTON ROD	2	H108
23	END MEMBER INNER NUT	1	H109
24	END MEMBER	1	H110
25	RAM NUT	1	H111

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY CALIF.
DWG. NO. C403
SECTION TOP VIEW
DRAWN BY: Harry J. Jameson
18 MARCH 1968



	PART	NO. REQ	DWG. NO.
1	BEARING ADJUSTER	2	B 201
2	TEST END BEARING HANGER	1	B 202
3	DRIVE END BEARING HANGER	1	B 203
4	BASE PLATE	1	B 204
5	TEST SECTION OIL PAN	1	L 301
6	DRIVE SECTION OIL PAN	1	L 302
7	TEST SECTION END	1	B 205
8	VIEW PORT	1	B 208
9	SHAFT	1	S 501
10	DRIVE GEAR	1	G 701
11	TEST GEAR	1	G 704
12	TOP SECTION	1	
13	ROLLER BEARING 28985 TDODC	4	
14	3/8" CAP SCREWS	20	
15	1/4" CAP SCREWS	38	

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. C402
SECTION SIDE VIEW
DRAWN BY: [signature]
7 MARCH 1966

2. Design Specifications.

The design criteria used in this project consisted of the following three specifications:

(1) The instrument must be capable of operation at speeds up to and including 10,000 RPM.

(2) The instrument must be capable of handling a total gear tooth load of 1000 pounds.

(3) The design must provide for start up under no load conditions and for loading to be accomplished while running.

3. Main Drive And Test Gear Shafts.

Because of the corrosion resistance and the higher strength properties of stainless steel, it was chosen as shaft material. With the exception of the spline area the maximum diameter of the shafts is two inches. The various shoulder diameters were dictated by bearing fit, seal clearances, gear hole size and drive motor shaft diameter. The dimensions are noted on drawing numbers S501 and S502 and reflect the choice of standard commercially available components. In choosing the shaft bearings, the largest size bearings, consistent with the two inch diameter limit, were used. This was done to permit the greatest loading with the least possible bearing deformation. Tapered roller bearings were chosen because of their reliability in carrying combined axial and radial loads for prolonged periods of time at high speeds.

The test gear shaft was made with a one half inch internal bore over its whole length. This was done to accommodate the leads from the torque measuring and the dynamic tooth load strain gages. The torque measuring strain gages which are part of the control system, are mounted on the test gear shaft; and the strain gage leads are brought into the shaft through radial holes. These radial holes are located sufficiently far from the gages so that they will not influence the gage readings in any way.

Both the motor coupling and the test section gears are attached to the shafts with shear pins. This was done to prevent damage to the unit in the event of an overload.

As a check on design the bearing loads, combined stress on the shaft, and the deflection of the test gear shaft were calculated. The maximum

shaft deflection is an important consideration because of the close tolerances placed on the shafts to affect a partial oil seal and to maintain the alignment of the test section gears. All calculations were made using the dimensions of the test gear shaft. The symbols defined below are used in the calculations for this section.

a	Distance from the center of the helical pinion to the center of the drive end bearing. $a = 3.50$ in.
B	Load on the drive end bearing caused by loading the helical pinion.
B'	Load on the test end bearing due to loading of the helical pinion.
b	Distance from the center of the helical pinion to the center of the test end bearing. $b = 14.40$ in.
C	Function of effective error and elasticity of gear materials. $C = 250$ /2/
D_i	Inner diameter of the test shaft. $D_i = 0.50$ in.
D_o	Outer diameter of the test gear shaft. $D_o = 1.50$ in.
E	Modulus of elasticity for steel. $E = 30{,}000{,}000$ psi.
F	Face width of test gear. $F = 0.375$ in.
F_1	Force required to maintain load on helical pinion.
J	Polar moment of inertia.
K_m	Numerical combined shock and fatigue factor to be applied to computed bending moment. $K_m = 1.5$ /2/
K_s	Numerical combined shock and fatigue factor to be applied to torsional moment. $K_s = 1.0$ /2/

K_t	Stress concentration factor.
L_s	Distance between the center of the test gear and the center of the test end bearing. $L_s = 2.56$ in.
M	Bending moment.
N_s	Factor of safety.
R	Radius of test gear shaft. R = 0.75 in.
R_p	Radius of gear pitch circle. $R_p = 2.90$ in.
T	Torque input to system.
v_m	Pitch line speed. $v_m = 2730$ ft/min.
W	Load on gear.
W_n	Design load on gear face. $W_n = 1000$ pounds.
W_d	Buckingham's dynamic load.
W_t	Tangential component of gear load.
α	Correction for equivalent column stress. $\alpha = 1.03$. /2/
δ	Deflection of shaft.
δ_d	Design maximum allowable deflection. $\delta_d = 0.001$ in.
τ	Shearing stress as a result of design load.
τ_d	Shear design stress. /2/ $\tau_d = 22{,}500$ psi.
σ	Bending stress as a result of design load.
σ_d	Normal design stress. /2/ $\sigma_d = 45{,}000$ psi.
ϕ	Tooth pressure angle. $\phi = 20$ degrees.

ψ Helix angle.
ψ = 12.6 degrees.

In figuring the effects of loading in the following calculations, the dynamic loading as proposed by Buckingham was used. This was done to allow for the various inaccuracies and accompanying accelerations that are always present. The dynamic loading is

$$W_d = W_n + \frac{0.05v_m(FC + W_n)}{0.05v_m + (FC + W_n)^{\frac{1}{2}}}$$

$$= (1000) + \frac{(.05)(2730)[(.375)(250) + (1000)]}{(.05)(2730) + [(.375)(250) + (1000)]^{\frac{1}{2}}}$$

$$= 1880 \text{ pounds.}$$

Using the value of dynamic load the tangential force and the applied torque are

$$W_t = W_n \cos\phi = (1880\#)(0.939) = 1765 \text{ pounds}$$

$$T = W_t R_p = (1765\#)(2.90\text{IN}) = 5120 \text{ pound inches.}$$

The torque imparted to the shafts by the test section gears is the same as that from the loading gears. Using this torque the force required to load the gears is

$$F_1 = W_t \tan\psi = (1765\#)(0.222) = 392 \text{ pounds.}$$

This also represents the maximum thrust load on the bearings. The maximum allowable thrust load is 990 pounds which produces a factor of safety of 3.96 in the axial direction. The radial bearing loads may be found by summing the moments around each bearing. Taking the sum of the moments around the test end bearing gives a drive end bearing radial load of

$$F_1 R_p - B(a + b) + W_d(b - L_s) = 0$$

$$B = \frac{F_1R_p + W_d(b - L_s)}{(a + b)}$$

$$= \frac{(392\#)(2.9IN) + (1880\#)(14.40IN - 2.56IN)}{(3.50IN + 14.40IN)}$$

$= 1300$ pounds.

The radial load on the test end bearing is

$$F_1R_p - W_da - W_d(a + b + L_s) + B'(a + b)$$

$$B' = \frac{W_d(2a + b + L_s) - F_1R_p}{(a + b)}$$

$$= \frac{(1880\#)(7IN + 14.4IN + 2.56IN) - (392\#)(2.9IN)}{(3.5IN + 14.4IN)}$$

$= 2440$ pounds.

This loading is higher than that recommended for the design speed. It must be remembered that the effect of loading above the limit set by the manufacturer is a reduction of bearing life. These particular bearings have an expected life of 1220 hours at 10,000 RPM. The life and load rating increase as operating speed is reduced. Since dynamic loading is not a steady phenomena and the value represents a maximum reached periodically, it would not be fair to evaluate bearing life from this factor alone. Also it is noted that Buckingham's equation is conservative and at reduced speeds the dynamic effects are less severe. In view of the loadings imposed, an extreme pressure lubricant will be used in the main lubrication system.

Having established that the bearing loads are within acceptable limits the next step is the determination of the maximum stress in the shaft. The maximum stress will occur just outside of the end bearing and consists of a torsional and bending moment component.

The moment of inertia at this point is

$$J = \frac{\pi(D_o^4 - D_i^4)}{32} = \frac{(3.1416)(1.5^4 - 0.5^4)}{(32)}$$

$$= 0.492 \text{ inches}^4.$$

The torsional shear stress is

$$\tau = \frac{K_t TR}{J} = \frac{(1.2)(5120\#IN)(0.75IN)}{(0.492\ IN^4)} = 9{,}400 \text{ psi.}$$

The bending moment is found from

$$M = W_d L_s = (1880\#)(2.56IN) = 4800 \text{ pound inches.}$$

This gives a bending stress of

$$\sigma = \frac{2MRK_t}{J} = \frac{(2)(4800\#IN)(0.75IN)(1.65)}{(0.492\ IN^4)} = 24{,}300 \text{ psi.}$$

When considering the effects of fatigue and reversed bending in the design of a rotating shaft the ASME code specifies the use of the maximum shear stress theory. Solving /2/

$$D_o^3 = \frac{16}{\tau_d \pi (1 - D_i^4/D_o^4)} \left[(K_s T)^2 + \left\{ K_m M + \frac{\alpha F_1 D_o (1 + D_i^2/D_o^2)}{8} \right\}^2 \right]^{\frac{1}{2}}$$

$$D_o^3 = \frac{(16)}{(22500)(3.1416)(1 - .5^4/D_o^4)} \left[(1.0^2)(5120^2) \right.$$

$$\left. + \left\{ (1.5)(4800) + \frac{(1.03)(392)D_o(1 + .5^2/D_o^2)}{(8)} \right\}^2 \right]^{\frac{1}{2}}$$

for the value of D_o yields a value of 1.27 inches. This is less than the dimension used so that an even greater margin of safety than the above theory gives is achieved.

The last item to check is the shaft deflection and verify that it is less than the allowable deflection of 0.001 inches. The deflection of the shaft is made up of three components and the solution will be done by superposition. First, the load acting between the two supports will

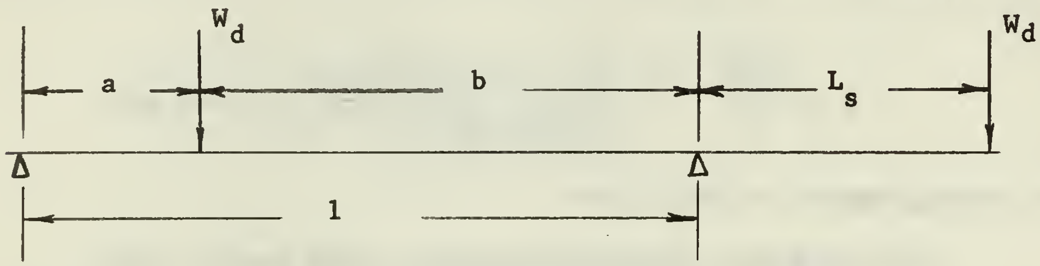


Figure 3.1

cause a slope at the right end.

$$\theta_1 = \frac{W_d a(1^2 - a^2)}{31EJ}$$

$$= \frac{(1880)(3.5)(17.9^2 - 3.5^2)}{(3)(17.9)(30 \times 10^6)(.492)}$$

$$= .00256 \text{ radians.}$$

This in turn leads to a deflection of

$$\delta_1 = L_s\theta_1 = .00655 \text{ inches which acts upward.}$$

The load on the right has two effects; the first is that of a tip load on a cantelever beam and is

$$\delta_2 = \frac{2W_d L_s^3}{3EJ} = \frac{(2)(1880)(2.56^3)}{(3)(30 \times 10^6)(.492)}$$

$$= 0.00152 \text{ inches.}$$

This load also produces a couple at the right hand support which gives an angle of

$$\Theta_2 = \frac{2W_d L_s l}{3EJ} = \frac{(2)(1880)(2.56)(17.9)}{(3)(30 \times 10^6)(.492)}$$

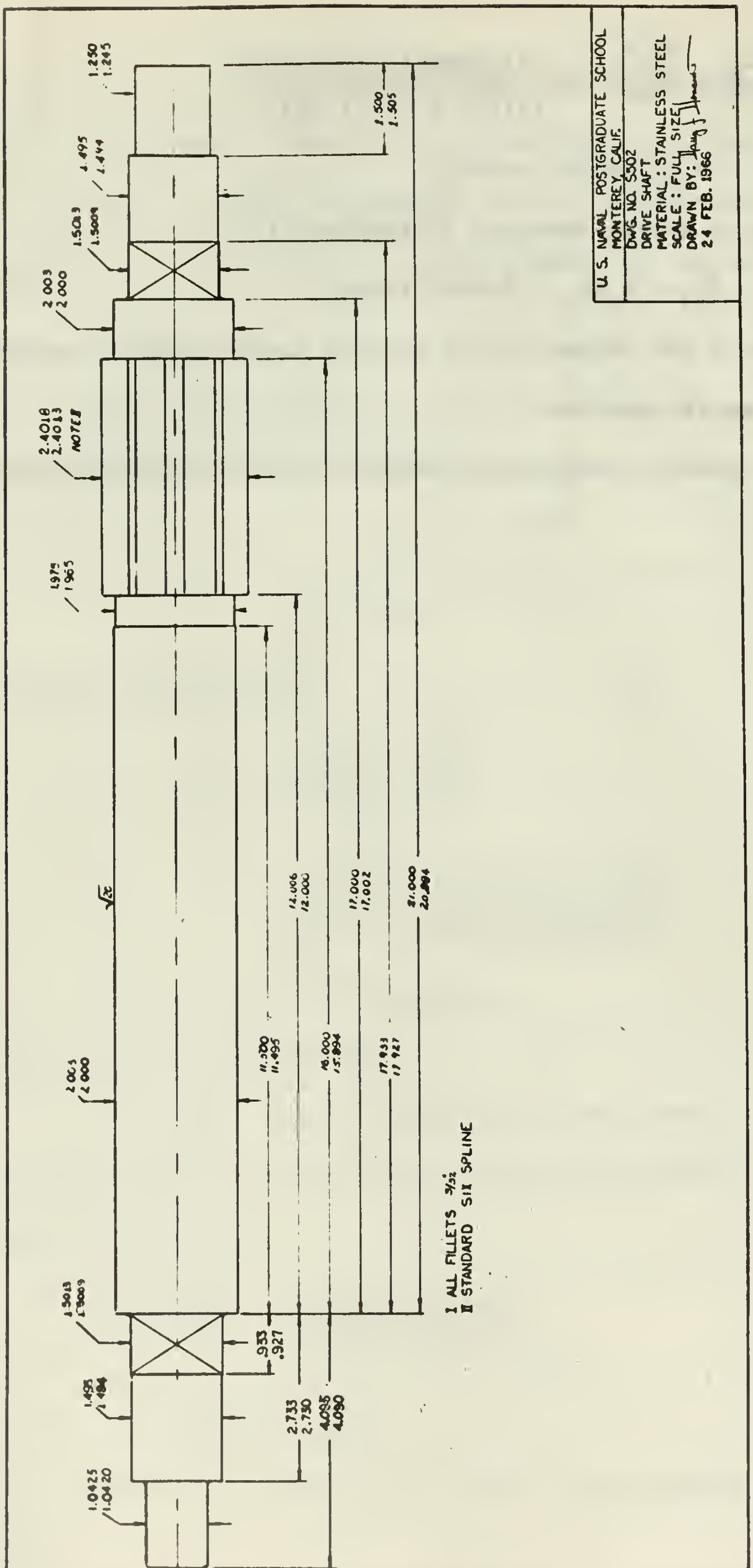
$$= 0.00190 \text{ radians.}$$

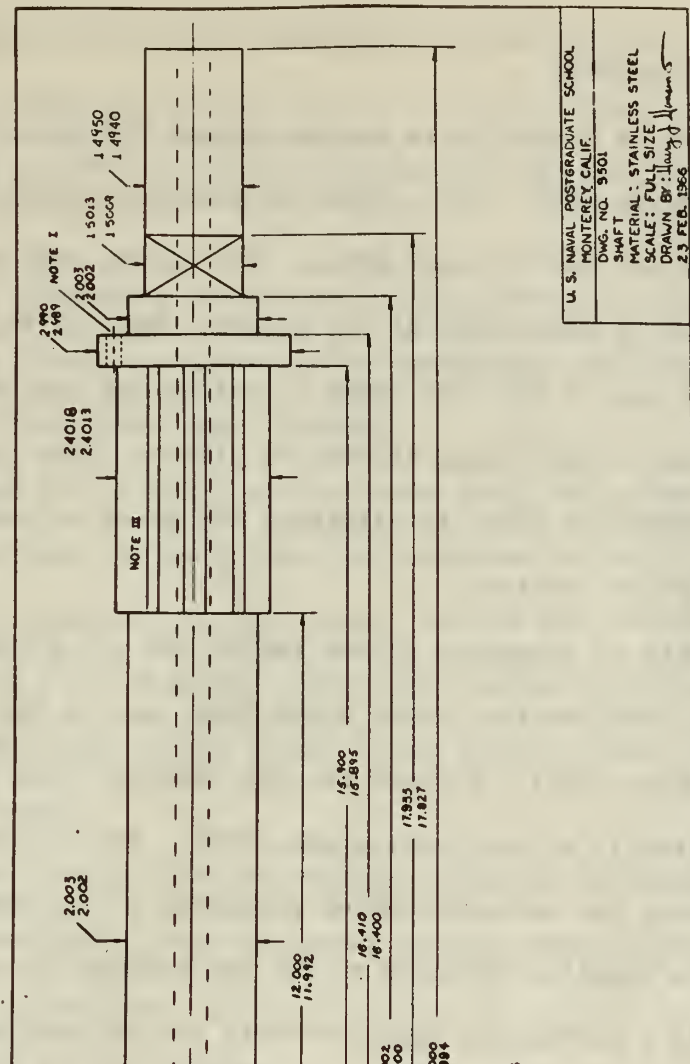
This leads to a second downward deflection of

$$\delta_3 = L_s \Theta_2 = 0.0049 \text{ inches}$$

This produces a net deflection of 0.00015 inches which is acceptable and the design is complete.

The two shaft drawings are presented on the following pages.


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. 5502
DRIVE SHAFT
MATERIAL: STAINLESS STEEL
SCALE: FULL SIZE
DRAWN BY:
24 FEB. 1966
I ALL FILLETS 3/32
II STANDARD SIX SPLINE
1.250
1.245
1.500
1.505
1.495
1.494
1.5013
1.5009
2.003
2.000
2.4018
2.4013
NOTE II
1.975
1.965
2.003
2.000
1.0425
1.0420
.933
.927
2.733
2.730
4.095
4.090
11.500
11.495
12.006
12.000
16.000
15.994
17.000
17.002
17.933
17.927
21.000



U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY CALIF.
DWG. NO. 5501
SHAFT
MATERIAL: STAINLESS STEEL
SCALE: FULL SIZE
DRAWN BY:
23 FEB. 1966
I 6 EQUALLY SPACED HOLES FOR ¼" SCREWS
II FOR ROLLER BEARING 27881
III STANDARD SIX SPLINE
IV ALL FILLETS 3/32"
V ½" HOLE ON SHAFT CENTERLINE
NOTE I
NOTE II
NOTE III
1.4950
1.4940
1.5013
1.5009
2.990
2.989
2.003
2.002
2.4018
2.4013
15.900
15.895
17.935
17.927
16.410
16.400
12.000
11.992
17.002
17.000
26.000
25.984
1.495
1.494
0.933
0.927
2.542
2.540
1.0425
1.0420
3.900
3.894

4. Hydraulic Assembly.

The machine is loaded by an annular shaped hydraulic cylinder that straddles the drive shaft. The system is somewhat unique in that the piston is fixed and the cylinder moves. The piston rods are utilized to supply the fluid to both sides of the piston. Note in the assembly drawing, C401, that one of the rods shown is hollow and that the other has a hole terminating in the flange before the piston. This system of fluid supply was employed in order to eliminate any hoses or moving fittings from the main drive section.

The assembly is supported on one end by the six piston rods which are attached to the bearing hanger block B202, and on the other end by the helical pinion, G702. Support is also realized from the ram, H101, which rides directly on the drive shaft, S501. The ram must move axially during loading and unloading which presented a lubrication problem. Although oil is supplied directly to the ram bearing there is no way of ascertaining if a sufficient amount of this oil is available to lubricate the ram. The solution was the application of a teflon coating to the inner surface of the ram that rides on the shaft.

When the design was started the machine was to have had a four inch center to center distance for the shafts, and the hydraulic assembly was designed accordingly. After most of the parts had been fabricated this distance was increased to 5.800 inches. The only benefit to be derived from a larger cylinder would be lower pressure requirements for the system. The pressure needs are not sufficient to warrant a change in existing hardware.

While not shown in the drawings, the ram nut, H111, and the end member nut, H109, are secured with set screws. These nuts allow for adjustments to be made to the ram bearing and simplify the assembly of the unit.

Table I presents a tabulation of the groove data for the "O" rings and provides an amplification of the dimensions and instructions set forth in the individual part drawings.

The inner wall, H105, and the outer wall, H104, were provided with a tapered entrance to facilitate the insertion of the piston and cover plate without damaging the "O" rings. It was not realized until after the inner wall and end member, H110, were welded together that they could have been made as one piece. This would, I feel, have resulted in a better design.

Using the design load the maximum pressure requirements for the system are developed. The following are the symbols used in the calculations for this section.

A_p Area of the piston.

A_r Reduced area of the piston due to the interference of the piston rods.

D_o Outer diameter of the piston.
$D_o = 4.87$ in.

D_s Inner diameter of the piston.
$D_s = 3.00$ in.

d Diameter of the piston rod.
$d = 0.498$ in.

F_1 Force required to move the helical pinion axially.

P Pressure required to attain desired loading.

W_t Tangential component of design load on helical pinion face.

W_n Design load on gears.
W_n = 1000 pounds.

ΣF_x Sum of the forces in the "x" direction.

ΣF_y Sum of the forces in the "y" direction.

μ Coefficient of friction.
μ = 0.15 /2/

ψ Helix angle.
ψ = 12.6 degrees

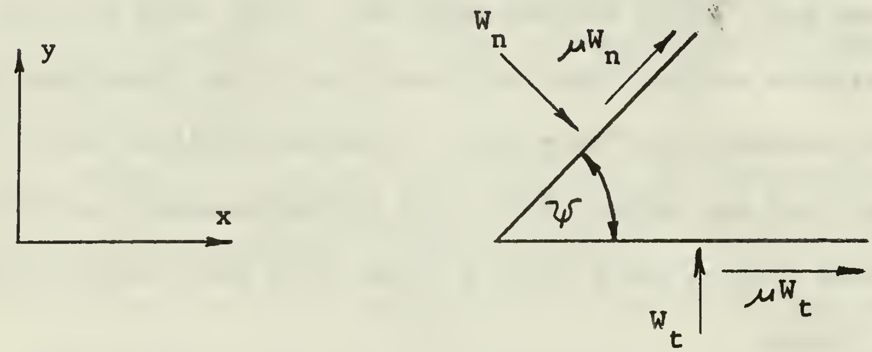


Figure 4.1

Using the above diagram as a reference, the axial force required to load the mechanism to the design point was determined. Summing the forces in the "y" direction gives

$$\Sigma F_y = 0$$

$$W_t + \mu W_n \sin\psi - W_n \cos\psi = 0$$

$$W_t = W_n(\cos\psi - \mu \sin\psi)$$

$$= (1000\#)(0.936 - (.15)(0.351)) = 884\#.$$

Taking the sum of the forces in the "x" direction and solving for the force F_1 results in a value of

$$\Sigma F_x = 0$$

$$F_1 - \mu W_t - \mu W_n \cos\psi - W_n \sin\psi = 0$$

$$F_1 = (0.15)(884) + (.15)(1000)(.936) + (1000)(.351)$$

$$= 624 \text{ pounds.}$$

Before calculating the required system pressure, the piston area and the reduced area due to rod interference must be determined. They are respectively

$$A_p = \frac{\pi(D_o^2 - D_s^2)}{4} = \frac{(3.14)(4.87^2 - 3.00^2)}{(4)}$$

$$= 11.58 \text{ in}^2.$$

$$A_r = A_p - \frac{6\pi d^2}{4} = 11.58 - \frac{(6)(3.14)(.498^2)}{(4)}$$

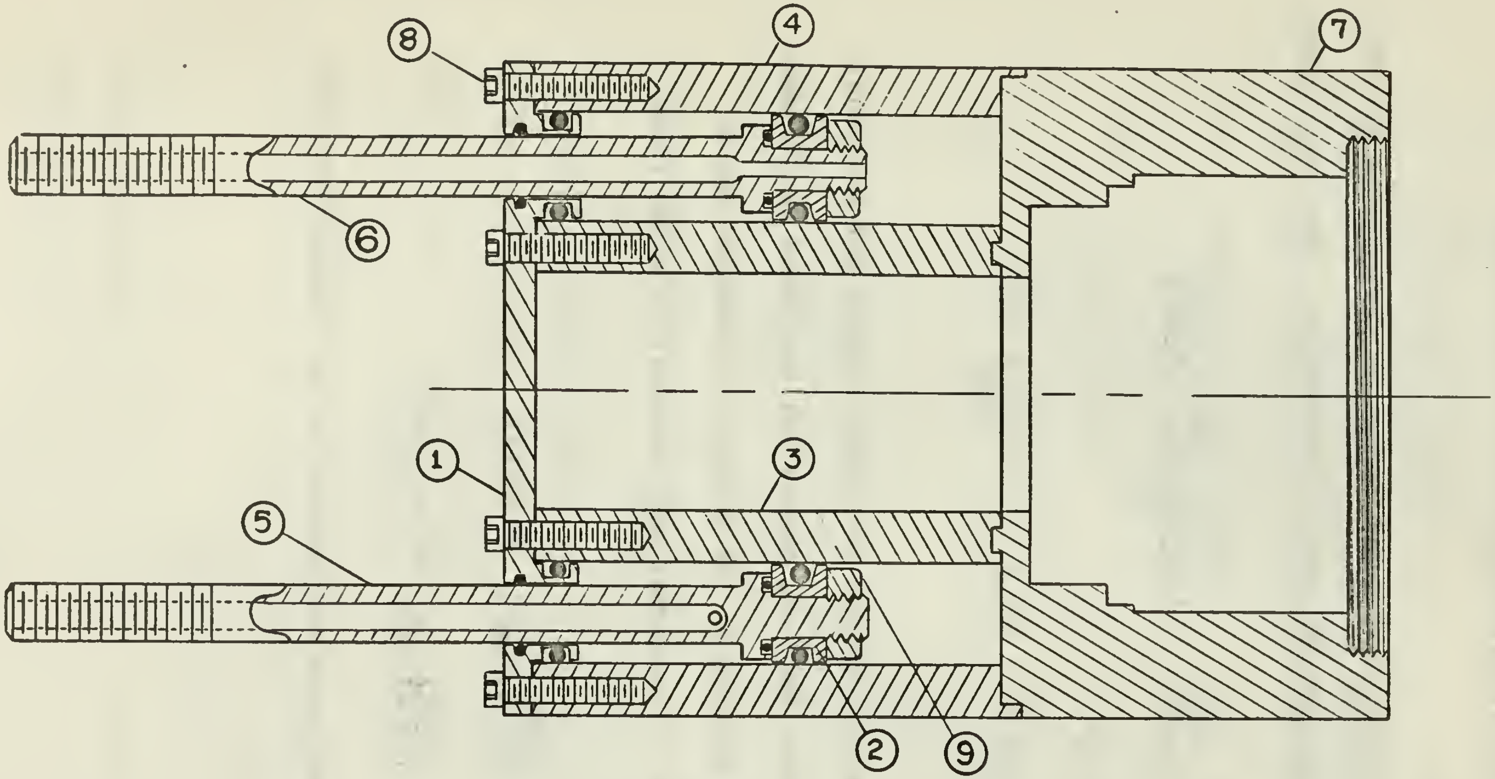
$$= 10.41 \text{ in}^2.$$

With these areas the pressure required to affect loading may be found. Two pressures were calculated because, as will be explained in the control and hydraulic system portion of the design, the machine may be loaded in either direction. The normal mode pressure is given first; the two pressures are

$$P = \frac{F}{A_p} = \frac{(624\#)}{(11.58 \text{ IN}^2)} = 54 \text{ psi.}$$

$$P = \frac{F}{A_r} = \frac{(624\#)}{(10.41 \text{ IN}^2)} = 59.9 \text{ psi.}$$

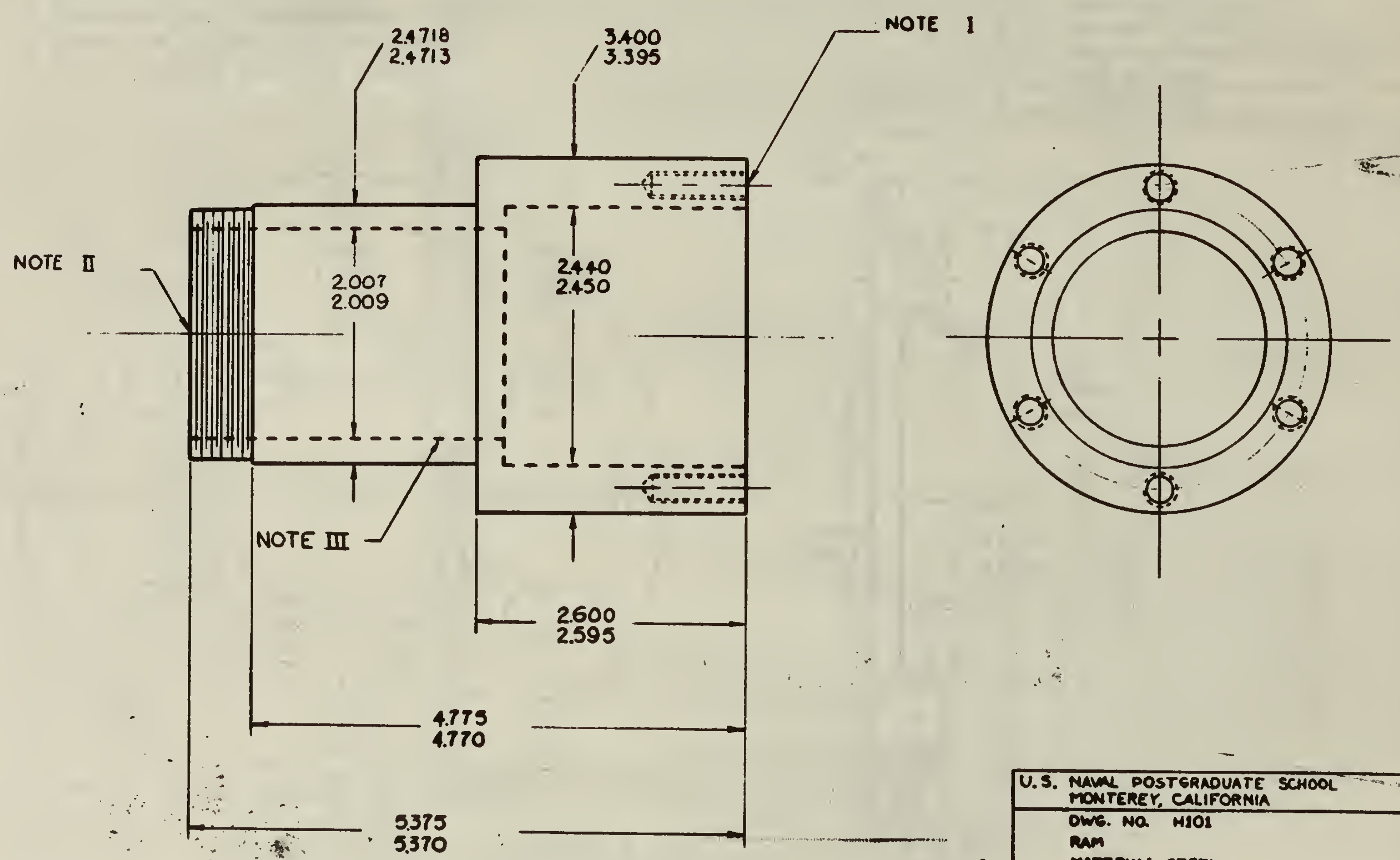
The following thirteen pages are devoted to the drawings for this section.



	NAME		DWG. NO.
1	END PLATE		H102
2	PISTON	(2)	H103
3	INNER WALL		H104
4	OUTER WALL		H105
5	PISTON ROD	(2)	H107
6	PISTON ROD	(2)	H108
7	END MEMBER		H110
8	1/4" CAP SCREW	(16)	
9	3/8" NUT	(6)	

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

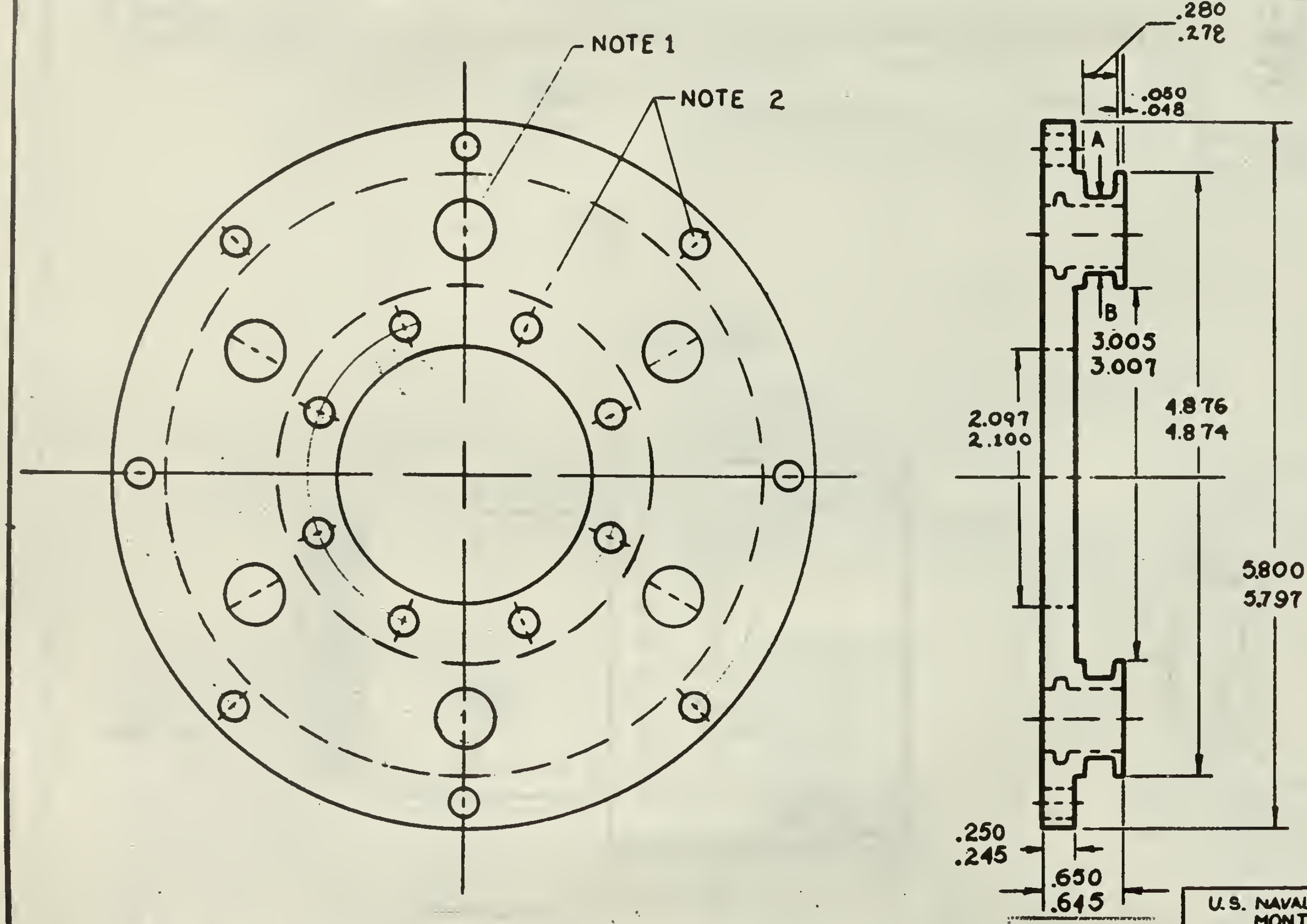
DWG. NO. C401
HYDRAULIC CYLINDER ASSEMBLY
SCALE : FULL SIZE
DRAWN BY : Harry J. [signature]
22 FEB. 1966



I 6 HOLES EQUALLY SPACED, 1" DEPTH, THREAD 1/4-28UNF-2A (HOLES TO MATCH GEAR)
II THREAD 2.4-20UNF-2A
III TEFLON COATING, FINAL I.D. 2.0005/2.0010

U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA

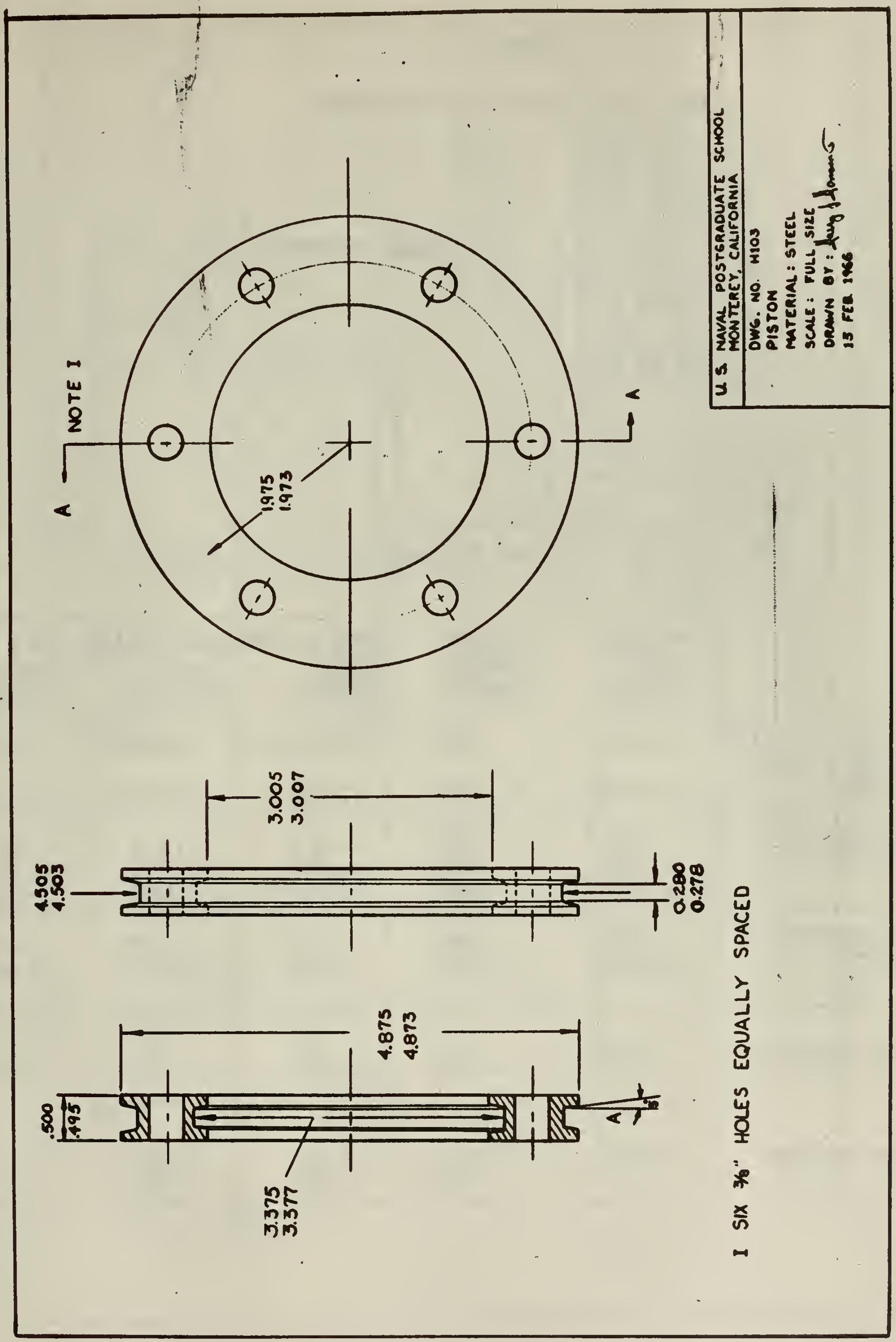
DWG. NO. H101
RAM
MATERIAL: STEEL
SCALE : FULL SIZE
DRAWN BY :
12 FEB. 1966



1 SIX ½" HOLES EQUALLY SPACED
2 EIGHT ¼" HOLES EQUALLY SPACED
3 DIMENSION "A" - 4.505/4.503
4 DIMENSION "B" - 3.375/3.377

U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

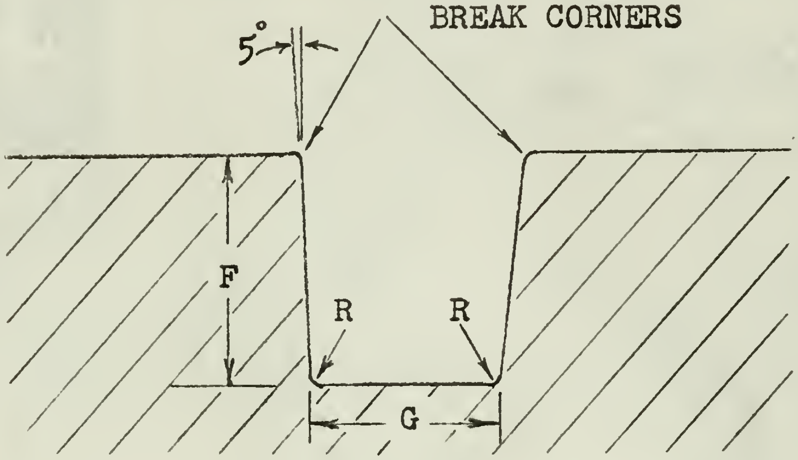
DWG. NO. H102
END PLATE, HYDRAULIC CYLINDER
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY :
12 FEB. 1966



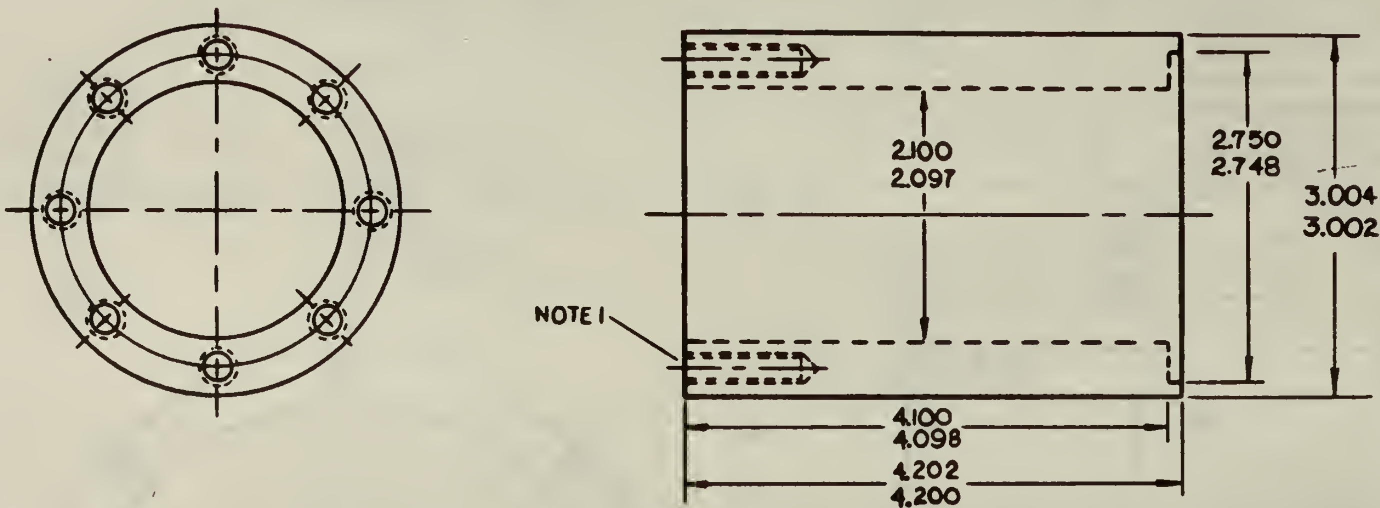
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
DWG. NO. H103
PISTON
MATERIAL: STEEL
SCALE: FULL SIZE
DRAWN BY:
13 FEB 1966
NOTE I
A
A
1.975
1.973
4.505
4.503
3.005
3.007
0.280
0.278
.500
.495
4.875
4.873
3.375
3.377
I SIX 3/8" HOLES EQUALLY SPACED

TABLE I

"O" RING GROOVE DIMENSIONS



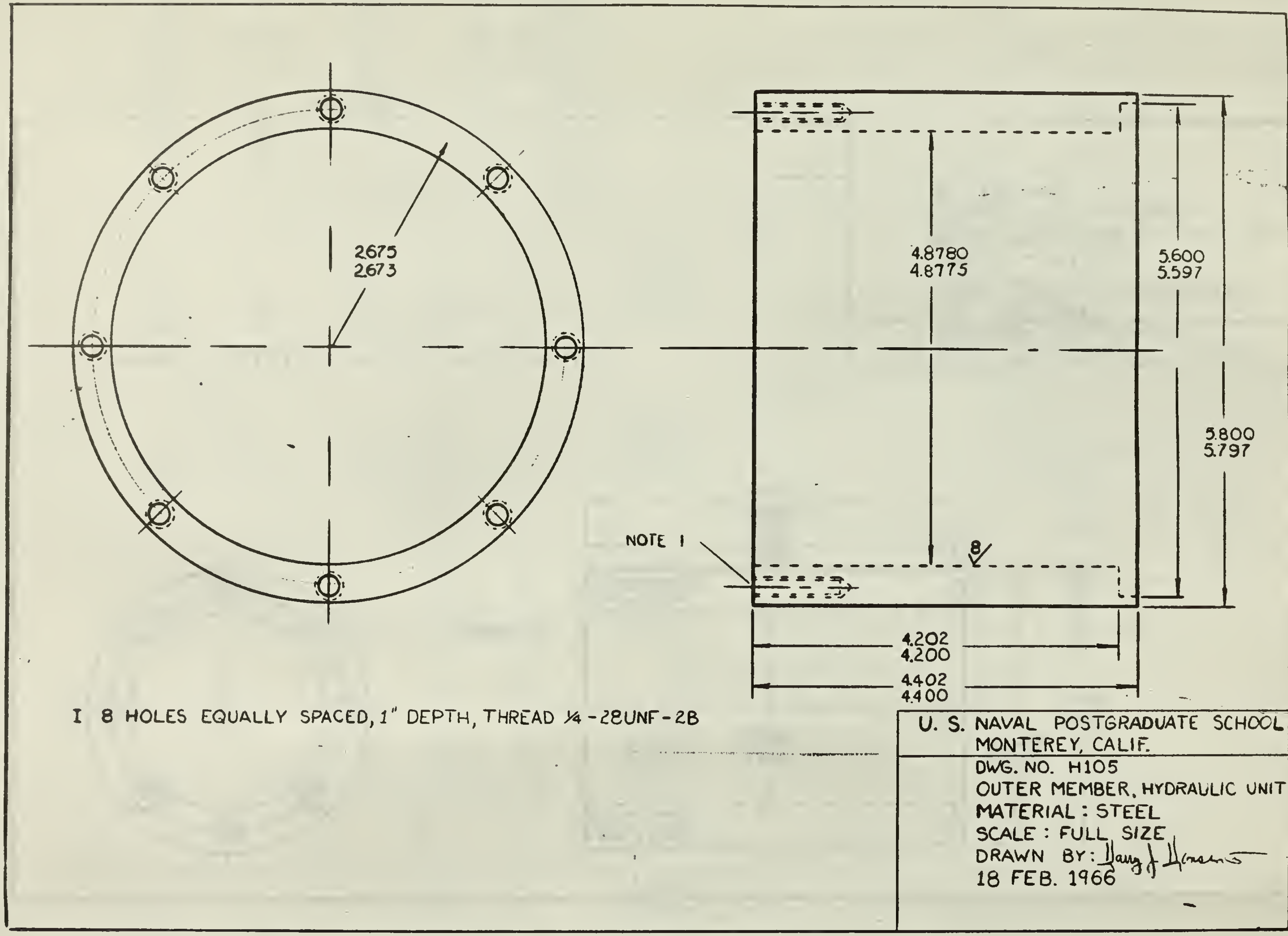
	PISTON OUTER GROOVE	PISTON INNER GROOVE	END PLATE OUTER GROOVE	END PLATE INNER GROOVE	END PLATE ROD GROOVE
DIAMETER "O" RING CROSS SECTION, W TOLERANCE	.210 ±.005	.210 ±.005	.210 ±.005	.210 ±.005	.103 ±.003
GROOVE DEPTH, F TOLERANCE	.188 +.000 −.001	.188 +.000 −.001	.173 +.000 −.001	.173 +.000 −.001	.090 +.000 −.001
GROOVE LENGTH, G TOLERANCE	.280 ±.005	.280 ±.005	.280 ±.005	.280 ±.005	.140 ±.005
MINIMUM RADIUS, R	.050	.050	.050	.050	.020
DRAWING NUMBER	H103	H103	H102	H102	H102

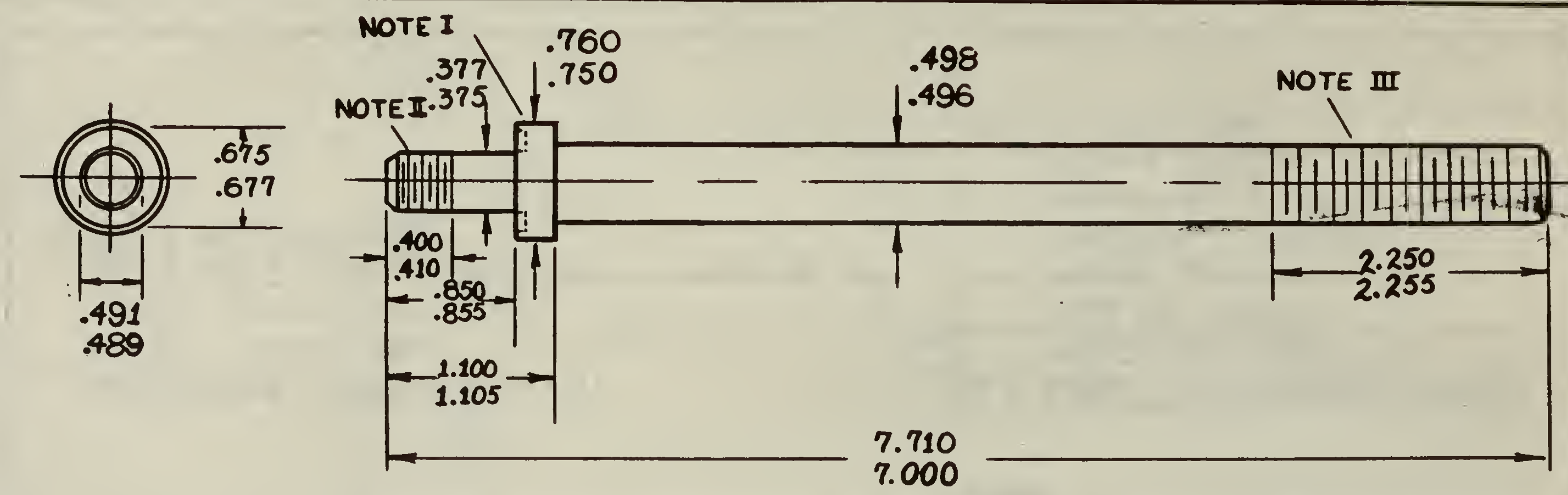


I 8 HOLES EQUALLY SPACED, ¼-28UNF-2B

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA

DWG. NO. H104
INNER MEMBER, HYDRAULIC UNIT
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY: Harry J. Hansen
16 FEB. 1966

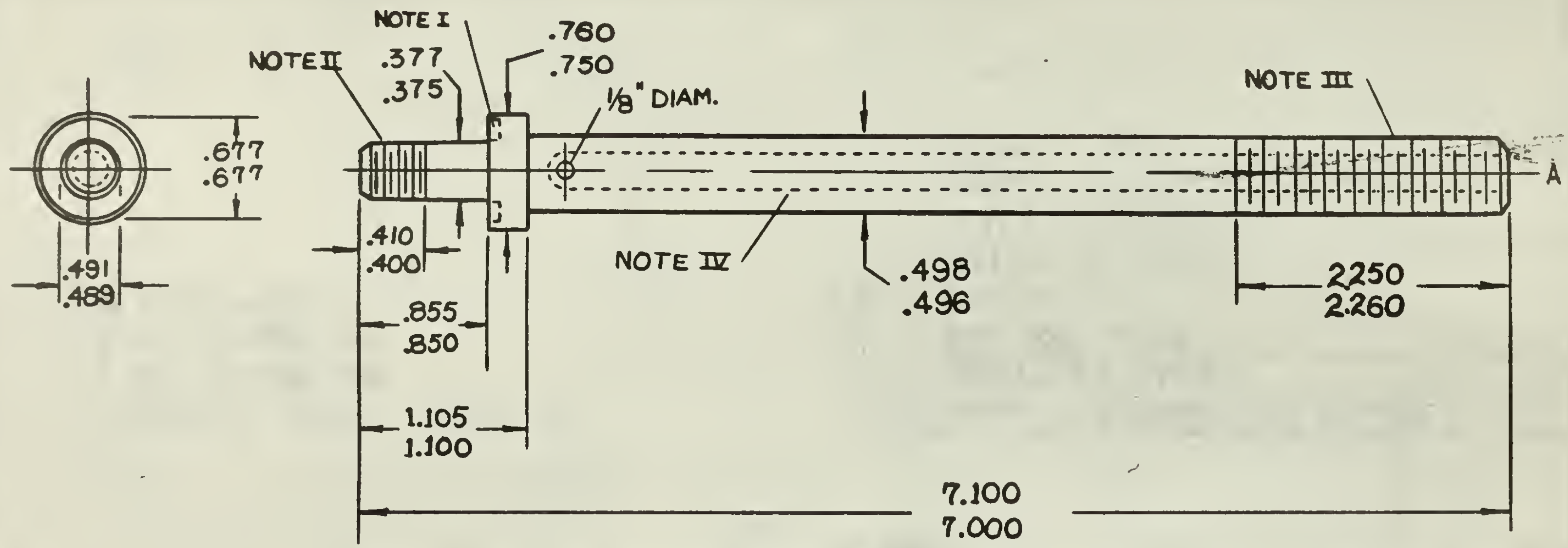

2675
2673
4.8780
4.8775
5.600
5.597
5.800
5.797
NOTE 1
8
4.202
4.200
4.402
4.400
I 8 HOLES EQUALLY SPACED, 1" DEPTH, THREAD 1/4 - 28UNF - 2B
U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. H105
OUTER MEMBER, HYDRAULIC UNIT
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY:
18 FEB. 1966



I GROOVE DEPTH .055/.057
II 3/8-24 UNF-2B
III 1/2-20 UNF-2B
IV MAKE TWO

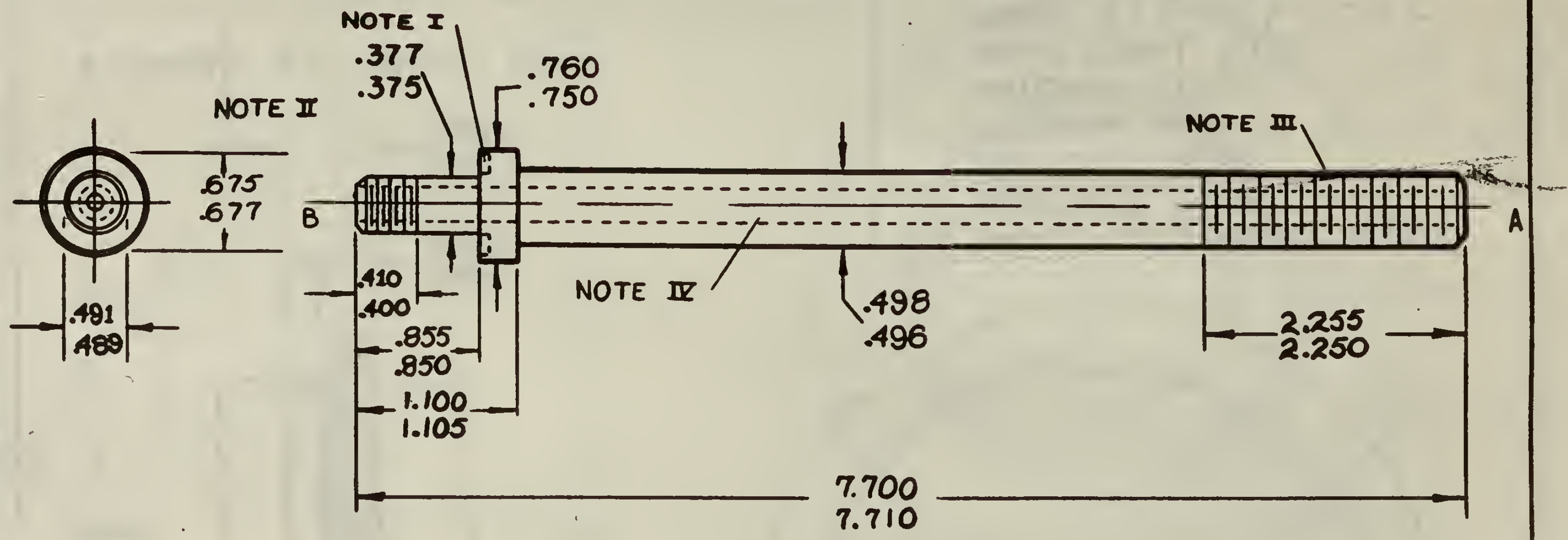
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. H106
PISTON ROD
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY : Harry J. Hansen
19 FEB. 1966



I GROOVE DEPTH .055/.057
II 3/8 - 24UNF - 2B
III 1/2 - 20UNF - 2B
IV 1/4" HOLE, LENGTH 6 1/4", A END THREAD 1/8" NP
V 2 REQUIRED

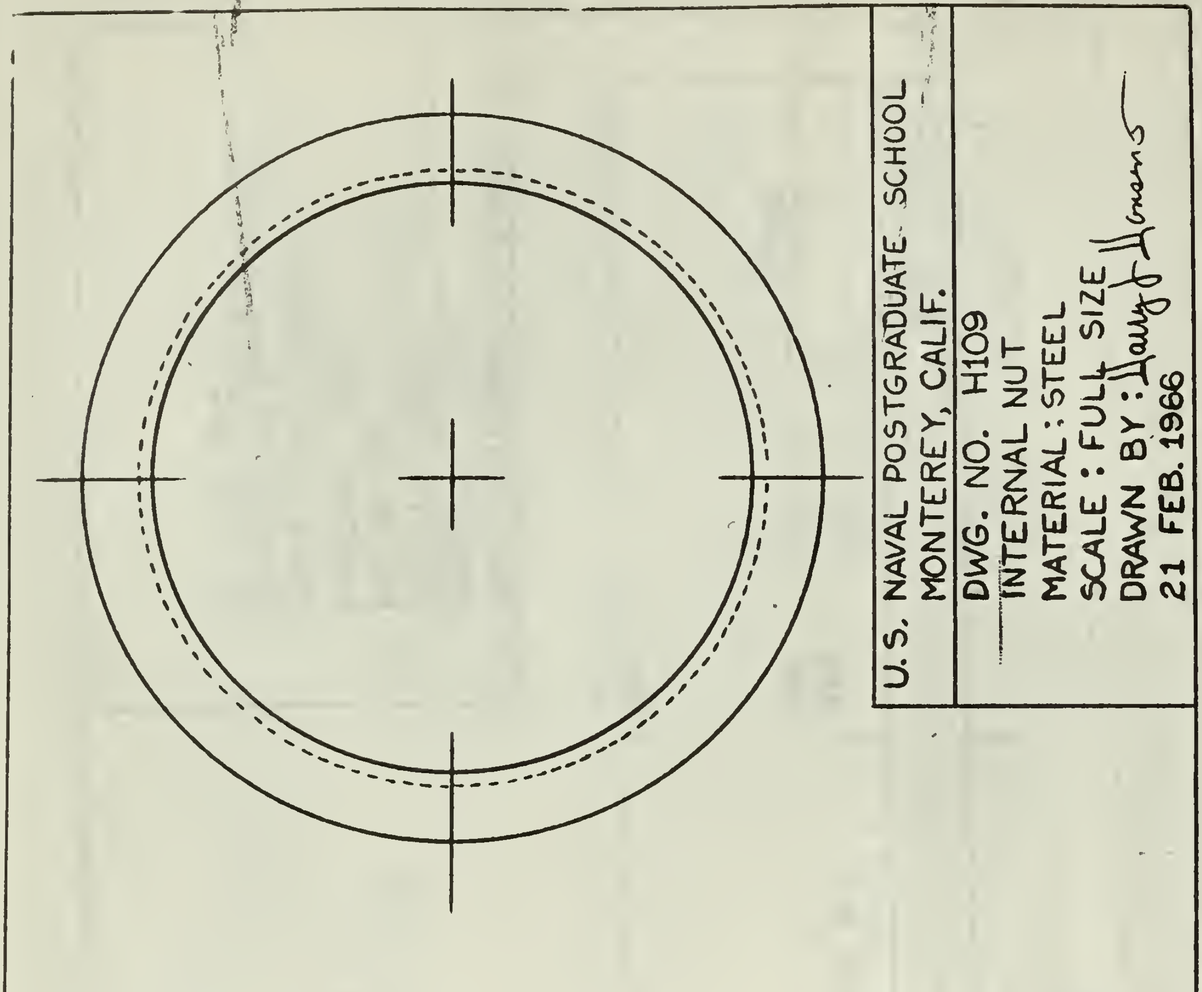
U. S. NAVAL POSTGRADUATE SCHOOL MONTEREY, CALIF.
DWG. NO. H107 PISTON ROD MATERIAL : STEEL SCALE : FULL SIZE DRAWN BY: Harry J. Hansen 21 FEB. 1966



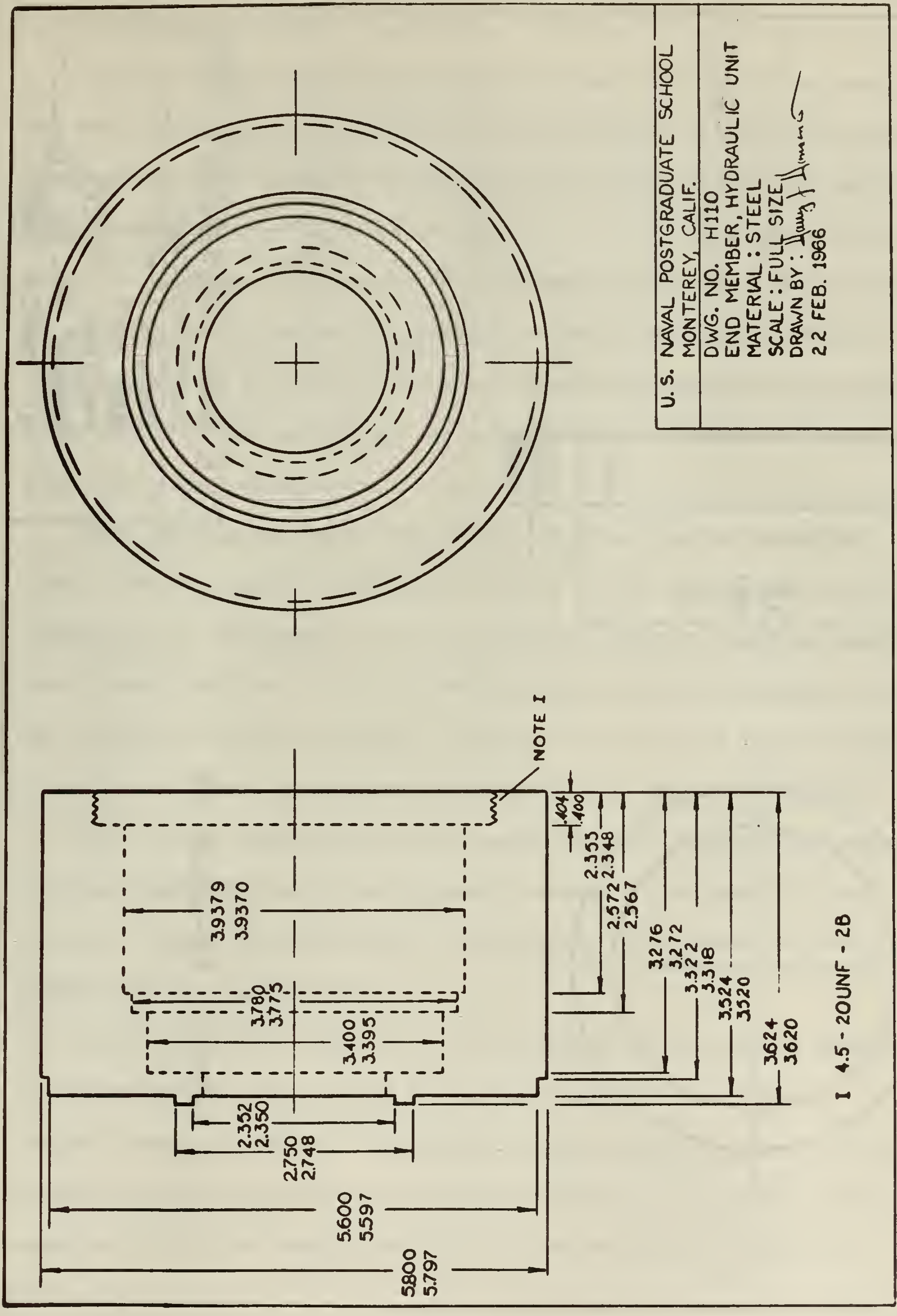
I GROOVE DEPTH .055/.057
II 3/8 -24UNF - 2B
III 1/2 -20UNF - 2B
IV 1/8" HOLE 2" FROM 'B' END, 1/4" DIAMETER REMAINDER, 'A' END THREAD 1/8" NP
V 2 REQUIRED

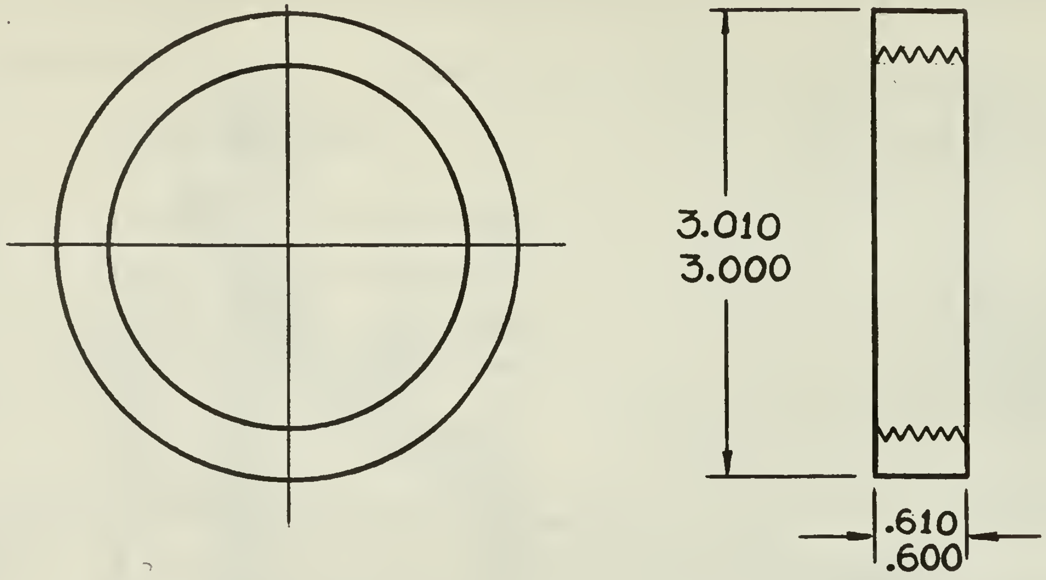
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. H108
PISTON ROD
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY: Harry J Hansen
21 FEB. 1966



U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. H109
INTERNAL NUT
MATERIAL: STEEL
SCALE: FULL SIZE
DRAWN BY: [signature]
21 FEB. 1966
3.790
3.780
0.400
0.396
0.622
0.616
3.937
3.934
4.500
4.492
I THREAD 4.5-20UNF-2B


U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. H110
END MEMBER, HYDRAULIC UNIT
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY :
22 FEB. 1966
NOTE I
I 4.5 - 20UNF - 2B
.404 .400
2.353 2.348
2.572 2.567
3.276 3.272
3.322 3.318
3.524 3.520
3.624 3.620
3.9379 3.9370
3.780 3.775
3.400 3.395
2.352 2.350
2.750 2.748
5.600 5.597
5.800 5.797



I THREAD 2.31-20UNF-2A

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. H111
RAM NUT
MATERIAL: STEEL
DRAWN BY: Harry J. Homan
17 MARCH 1968

5. Lubrication Systems.

There are two separate lubrication systems provided in the design. The main system serves the helical gears, and the five tapered roller bearings; the test system provides lubricant solely to the test gears. Both heaters and an oil cooler are utilized. The heaters are provided mainly to heat the fluid to the testing temperature before the machine is started. They also aid in keeping the fluid temperature constant while testing is in progress. The heat exchangers remove the heat gained by the fluid in lubricating the gears and bearings and thus maintain a uniform supply temperature.

Both systems are thermostatically controlled over an operating range of 80°F to 250°F. The upper limit is set by the maximum operating temperature of the pumps. If it is desired to conduct tests at temperatures above 250°F heaters may be added between the pump discharge and the nozzles in the test section. The main drive section was thermostated so that it could be used as an auxiliary test section if desired.

Most of the lubricating system components are located on a lower platform about two feet below the main assembly. The pumps and sump tanks are easily accessible on a sliding plate which forms the lower platform of the base structure.

The fluid heaters which are of the emersion type, and the thermostat sensing bulbs are located in the sump tanks of each system. Iverson recommends that the sumps be of large capacity so that an even temperature distribution may be achieved without an elaborate control network. /10/ The large capacity will also be needed in the main

system when a step up in speed is attempted. This is because the gear box to step up the drive shaft speed is lubricated by the main system. Ryder states that sufficient oil to carry away 60,000 BTU/HR is required when operating at 10,000 RPM with a drive power of from forty to fifty horsepower. /13/ Because of the future needs of the cooling system, the heat exchangers that are installed have a greater capacity than that required for the initial phase of operation. /7/

The system design is not complex yet affords a good deal of flexability. During the lubricant warm up period the fluid may be circulated in the bypass lines which will ensure flow around the heaters and provide for more even heating in the tank. Flow to the machine may be regulated precisely by the two needle valves located in the main and bypass lines. A relief valve is included as a safety feature to protect the system in the event of component failure or human error in operating the controls.

The system constants are tabulated in Table II, and the system parameters are listed in Table III. The following symbols are used in the calculations for this section. The values given with the symbols will be for the main lubrication system.

A Required surface area for cooling.

A_T Surface area of sump tank.

C Capacity of sump tank.

$C_{p_{SS}}$ Specific heat of stainless steel. $C_{p_{SS}} = 0.110$ BTU/#°F /3/

$C_{p_{Oil}}$ Specific heat of lubricating oil. $C_{p_{Oil}} = 0.509$ BTU/#°F /3/

E Power absorbed by lubricant.
$E = 4.50$ HP /13/

h Height of sump tank.
$h = 6.0$ in.

h' Height of fluid in tank.
$h' = 5.0$ in.

LMTD Log Mean Temperature Difference.
LMTD = 112 /7/

L_S Heat loss from surface of sump tank.
$L_S = 406$ watt hours. /11/

l Length of sump tank.
$l = 23.0$ in.

Q Killowatt hours required for initial heating of lubricating fluid.

Q_s Heater power.
$Q_s = 5000$ watts. /11/

T Time to heat fluid to operating temperature.

t Thickness of tank material.
$t = 0.062$ in.

U_W Overall heat transfer coefficient.
$U_W = 100$ BTU/HR FT^2°F /7/

V_{SS} Volume of the metal making up sump tank.

V Liquid volume of sump tank.

w Width of the sump tank.
$w = 10.0$ in.

w_{SS} Weight of the sump tank.

w_{Oil} Weight of lubricating fluid in tank.

$\dot{w}_{Oil}$ Flow rate of lubricating oil.
$w_{Oil} = 1.0$ GPM /8/

$\dot{w}_W$ Flow rate of water through heat exchanger.
$w_W = 4.0$ GPM /7/

ΔT_H Change in temperature between ambient and operating conditions.

ΔT_L Change in the temperature of the lubricating fluid passing through the heat exchanger.

ΔT_W Change in the temperature of the water passing through the heat exchanger.

ρ Density of lubricating fluid.
ρ = 53.19 #/ft^3. /3/

ρ_{SS} Density of the stainless steel tank material.
ρ_{SS} = 480 #/ft^3 /15/

As a preliminary to the heater requirements and the heat exchanger calculations, the sump tank volume, capacity, weight, and surface area plus the weight of the fluid in tank must be computed. They are respectively

$$V = \frac{whl}{1728} = \frac{(10\ IN)(6\ IN)(23\ IN)}{(1728\ IN^3/FT^3)} = .800\ ft^3$$

$$C = \frac{7.48whl}{1728} = \frac{(7.481\ GAL/FT^3)(10\ IN)(5\ IN)(23\ IN)}{(1728\ IN^3/FT^3)}$$

$$= 5.0 \text{ gallons}$$

$$W_{SS} = \frac{2\rho_{SS}t}{1728}\left\{wh + wl + hl\right\}$$

$$= \frac{(2)(480\#/FT^3)(.062IN)}{(1728\ IN^3/FT^3)}\left\{(10)(6) + (10)(23) + (6)(23)\right\}$$

$$= 16.0 \text{ pounds}$$

$$A_T = \frac{[2wh + 2lh + lw]}{144}$$

$$= \frac{(2)(10IN)(6IN) + (2)(23IN)(6IN) + (10IN)(23IN)}{(144\ IN^2/FT^2)}$$

$$= 4.36\ ft^2$$

$$w_{Oil} = \frac{C \rho_{Oil}}{7.481} = \frac{(5.0 \text{ GAL})(53.19\#/FT^3)}{(7.481 \text{ GAL}/FT^3)} = 36.2 \text{ pounds.}$$

To determine the energy required to heat the fluid an operating temperature of 176°F and an average ambient temperature of 70°F were assumed. This results in a ΔT_H of 106°F and the kilowatt hours for initial heating are /11/

$$Q = \frac{w_{Oil} C_{p\ Oil} \Delta T_H + w_{SS} C_{pSS} \Delta T_H}{3412} + \frac{L_S}{1000}$$

$$= \frac{(36.2)(.479)(106) + (16.0)(.110)(106)}{(3412)} + \frac{(406)}{(1000)}$$

$$= 1.001 \text{ KW HRS.}$$

From this value a heater was selected that would heat the fluid in less than thirty minutes. This time was considered the maximum allowable, and corresponds roughly to the time it might take to prepare for an experimental run. By choosing one 2.5 kilowatt heater the fluid will be heated in

$$T = \frac{60Q}{Q_S} = \frac{(60 \text{ MIN/HR})(1.001 \text{ KW HRS})}{(2.5 \text{ KW})} = 24 \text{ min.}$$

It is noted at this point that the test system time is shorter even though ΔT_H was greater. Twice as much power was used to heat the test fluid which more than compensated for the higher ΔT_H. The greater temperature difference stemmed from using the maximum allowable temperature of 250°F as the operating point in figuring ΔT_H.

The heat exchanger calculations were made assuming that the input power of 7.5 horsepower would be given up as heat to the oil. It was further assumed that the main section would absorb 4.5 horsepower and

that the remainder would be absorbed by the test system. The heat exchangers that were selected have a higher capacity than is needed initially. This is in anticipation of future needs and means only that the water flow rate must be increased when the input power is increased.

The first step is the determination of the temperature rise in the oil in the main assembly. This will then be the number of degrees that must be dropped across the exchanger. The temperature drop is

$$\Delta T_L = \frac{E}{\dot{W}_{Oil} C_{P_{Oil}}}$$

$$= \frac{(4.5HP)(7.48\ GAL/FT^3)(2545BTU/HP\ HR)}{(1GAL/MIN)(60MIN/HR)(.479BTU/\#\ F)(53.19\#/FT^3)}$$

$$= 55°F.$$

Assuming a cooling water flow rate of four gallons per minute, which is the minimum flow rate for the type of heat exchanger used, gives a change in water temperature of /7/

$$\Delta T_W = \frac{E}{500\ \dot{w}}$$

$$= \frac{(4.5HP)(2545BTU/HP\ HR)}{(500)(4.0GAL/MIN)} = 6.0°F.$$

With these two temperature differences the LMTD for a counterflow heat exchanger may be determined from tabulated values. /7/ With this value of 112 the area required may be found and a suitable heat exchanger chosen. The required heat transfer surface is

$$A = \frac{E}{(LMTD)U_W} = \frac{(4.5HP)(2545BTU/HP\ HR)}{(112)(100BTU/HR\ FT^2{}°F)}$$

$$= 1.02ft^2.$$

The lubrication system data, block diagram, and component drawings are presented on the next eleven pages.

TABLE II

CONSTANTS USED IN LUBRICATION SYSTEM DESIGN

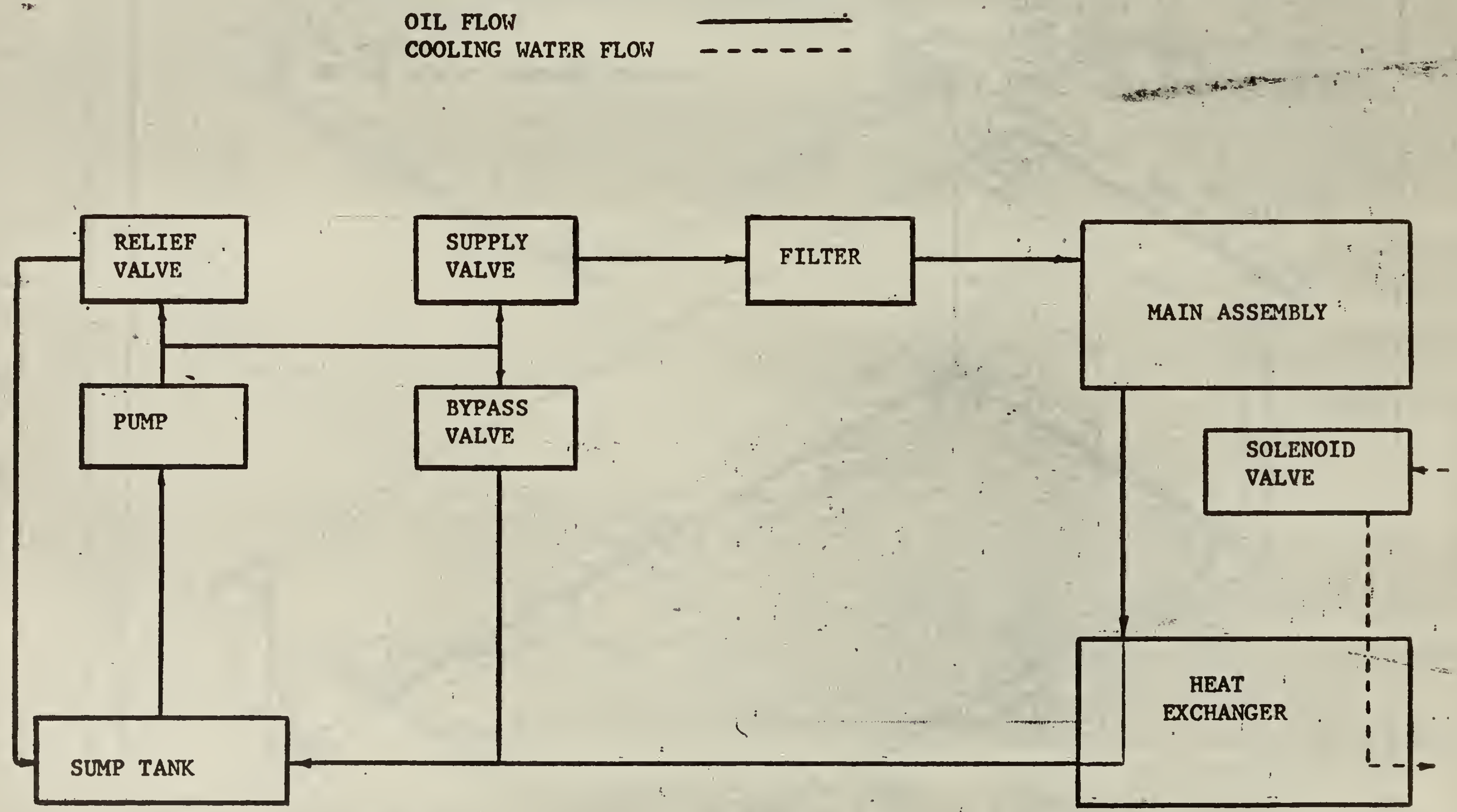
	MAIN SYSTEM	TEST SYSTEM	UNITS
A_T	4.360	4.360	FT^2
C	5.000	5.000	GALLONS
C_P	0.110	0.110	BTU/#°F
$C_{P_{Oil}}$	0.479	0.479	BTU/#°F
E	4.500	3.000	HP
$\dot{w}_W$	4.000	4.000	GPM
$\dot{w}_{Oil}$	1.000	0.600	GPM
L_S	436	436	WATT HOURS
Q_S	2500	5000	WATTS
ΔT_H	106	195	°F
U_W	275	275	$BTU/HR\text{-}FT^2 {}^\circ F$
V	0.800	0.800	FT^3
V_{SS}	0.036	0.036	FT^3
w_{SS}	16.0	16.0	POUNDS
w_{Oil}	36.2	36.2	POUNDS

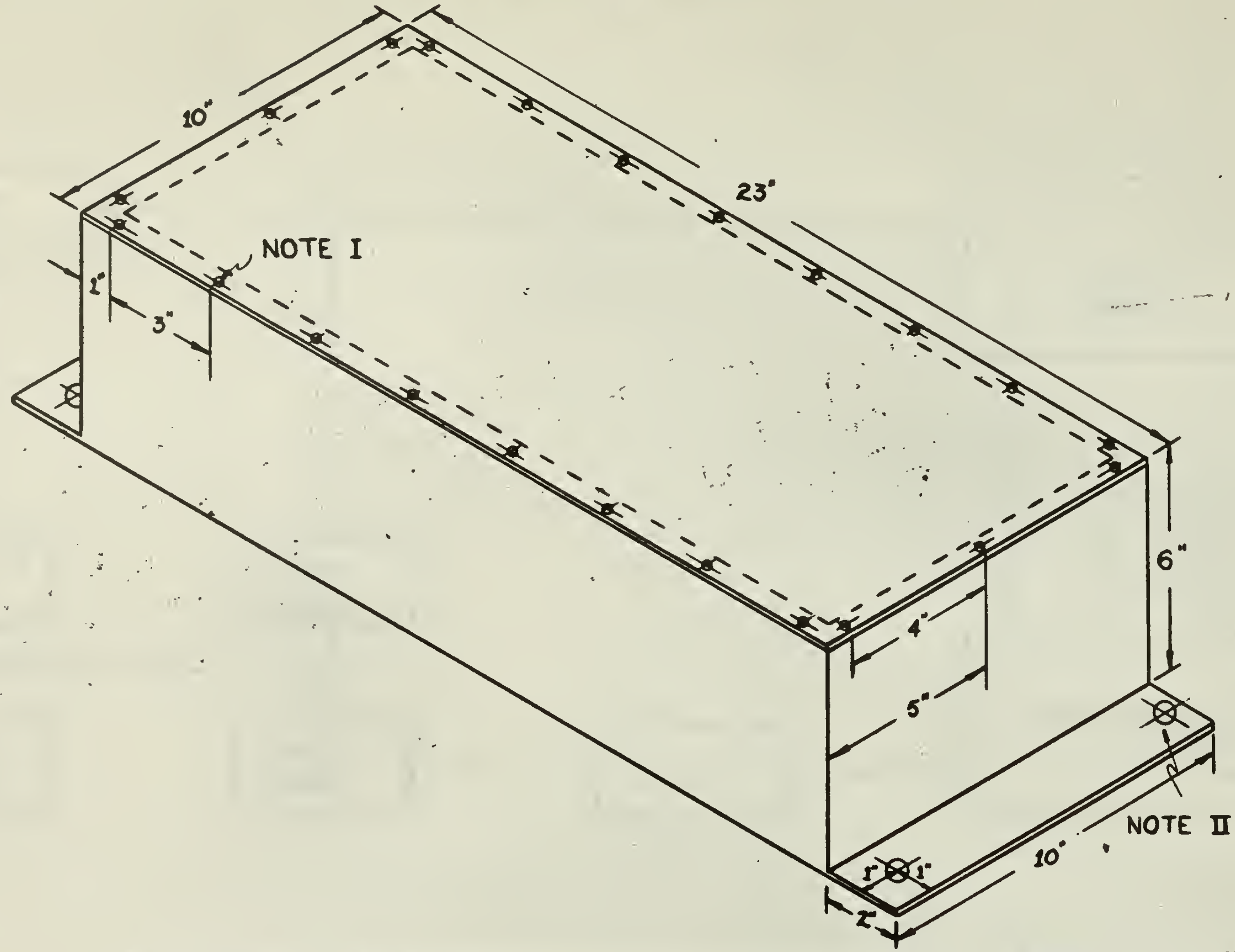
TABLE III

LUBRICATION SYSTEM PARAMETERS

	MAIN SYSTEM	TEST SYSTEM	UNITS
A	3.70	2.50	FT^2
LMTD	112	115	-
T_S	24	18	MINUTES
Q	1.029	1.576	KWATT HRS
ΔT_L	55.0	60.0	°F
ΔT_W	6.0	4.0	°F

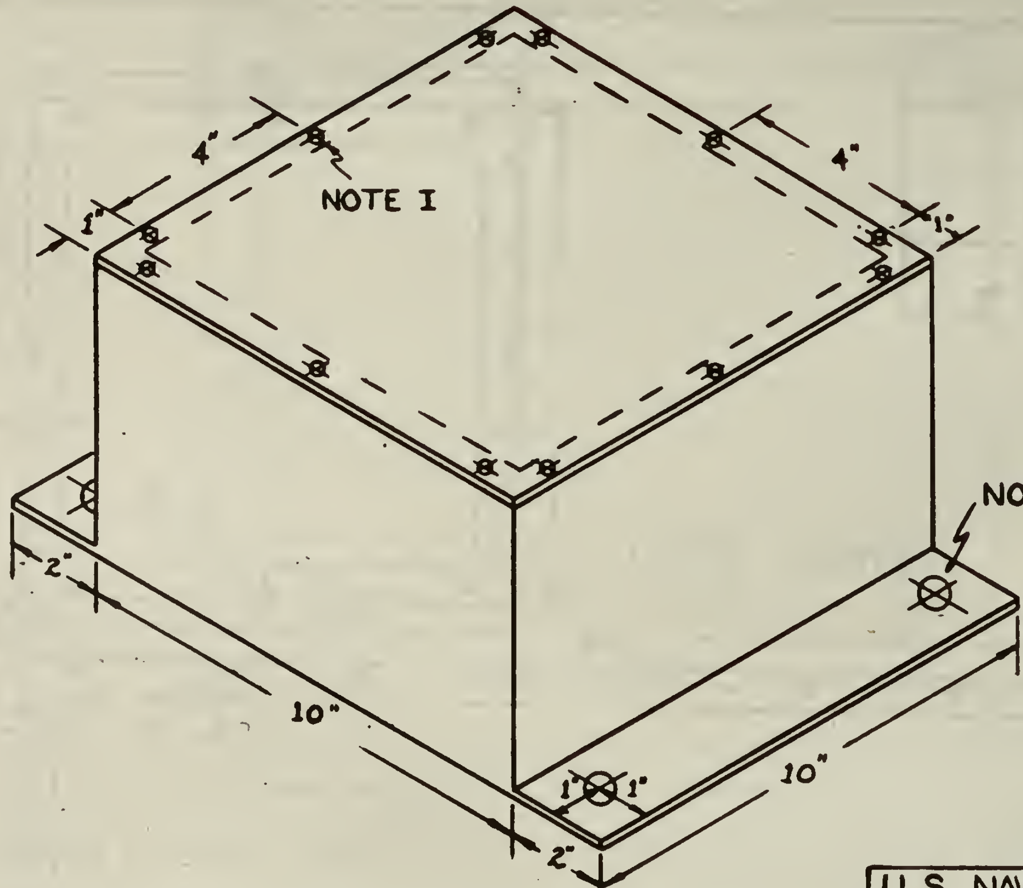
LUBRICATION SYSTEM BLOCK DIAGRAM





I HOLES FOR 1/4" MACHINE SCREWS
II 1/2" HOLES

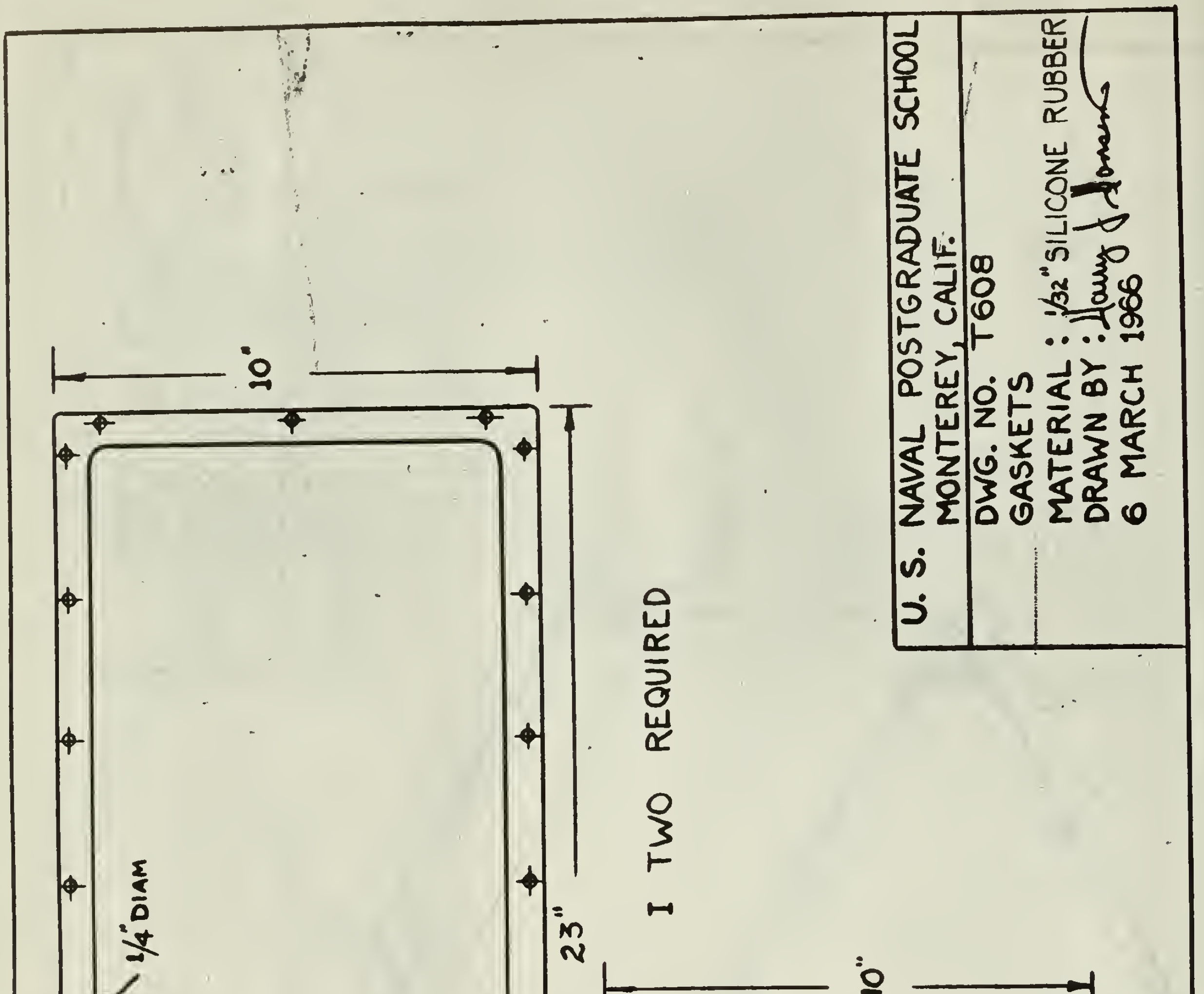
U. S.	NAVAL POSTGRADUATE SCHOOL MONTEREY, CALIF.
	DWG. NO. T601 OIL TANK MATERIAL: STAINLESS STEEL SCALE: ONE THIRD DRAWN BY: [signature] 28 FEB. 1966

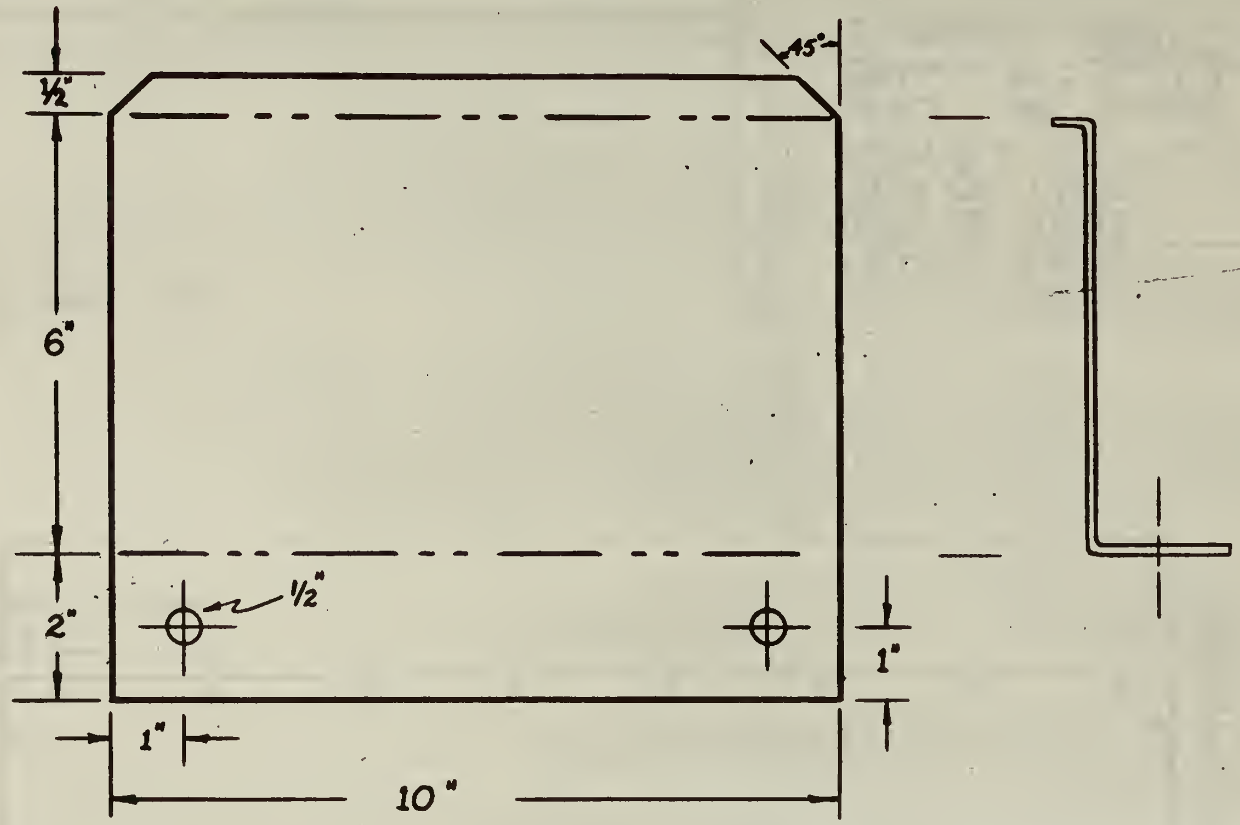


I HOLES FOR ¼" MACHINE SCREWS
II ½" HOLES

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. T607
HYDRAULIC FLUID TANK
MATERIAL: STAINLESS STEEL
SCALE: ONE THIRD
DRAWN BY:
1 MARCH 1966

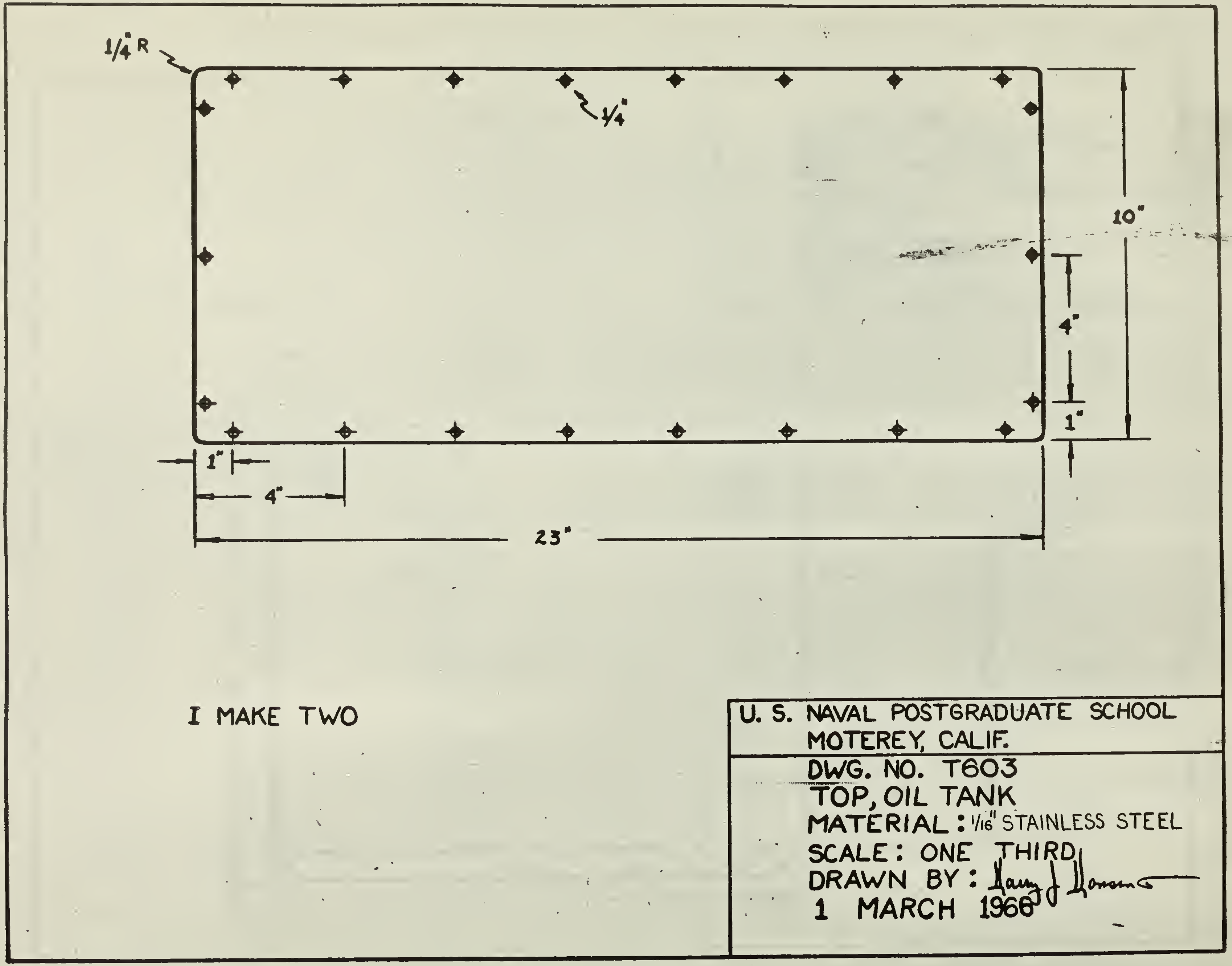

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. T608
GASKETS
MATERIAL : 1/32" SILICONE RUBBER
DRAWN BY :
6 MARCH 1966
I TWO REQUIRED
NOTE I
1/4" DIAM
1/2" R
1/4" R
10"
23"
4"
1"
1/4"
3/4"



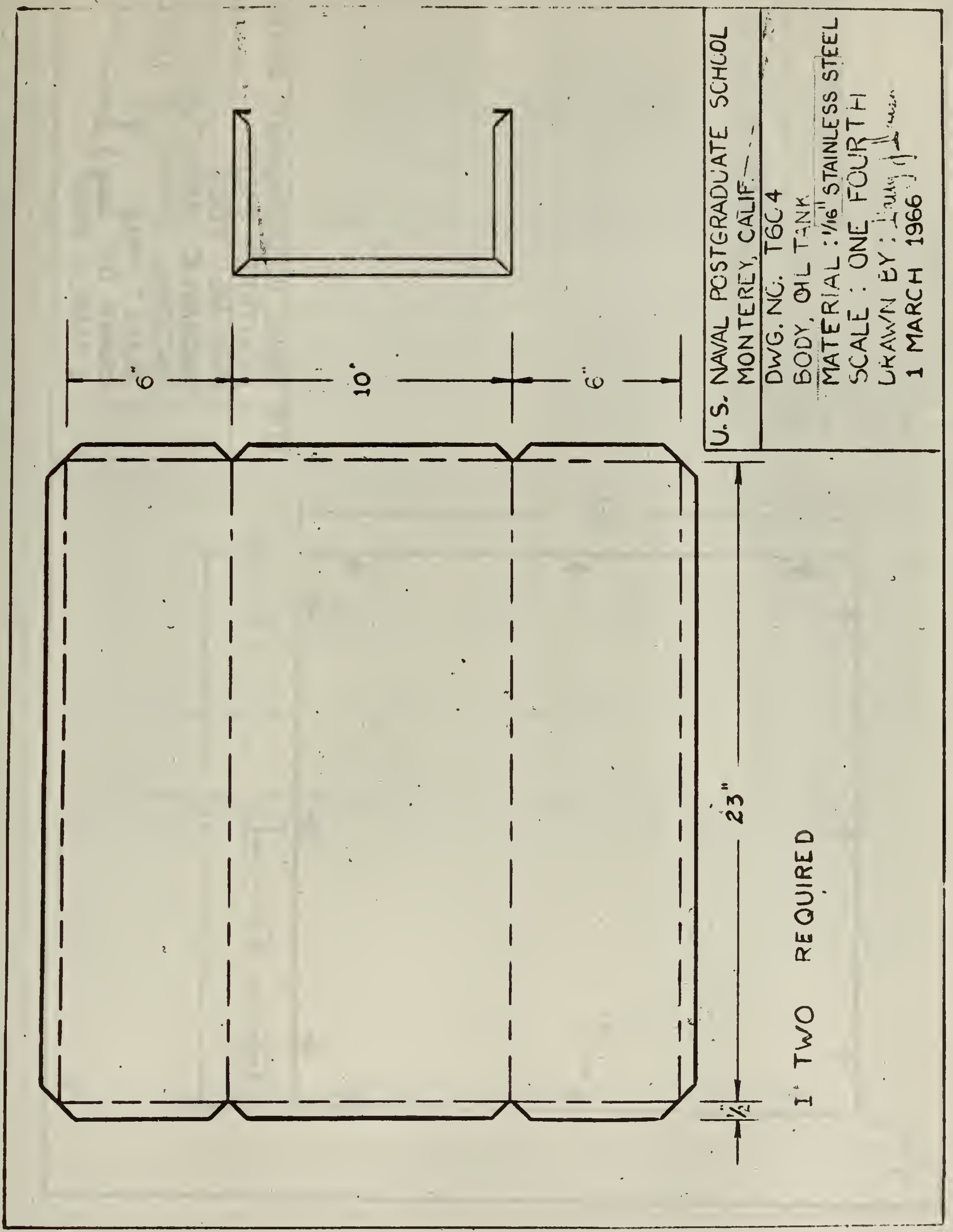
I MAKE SIX

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

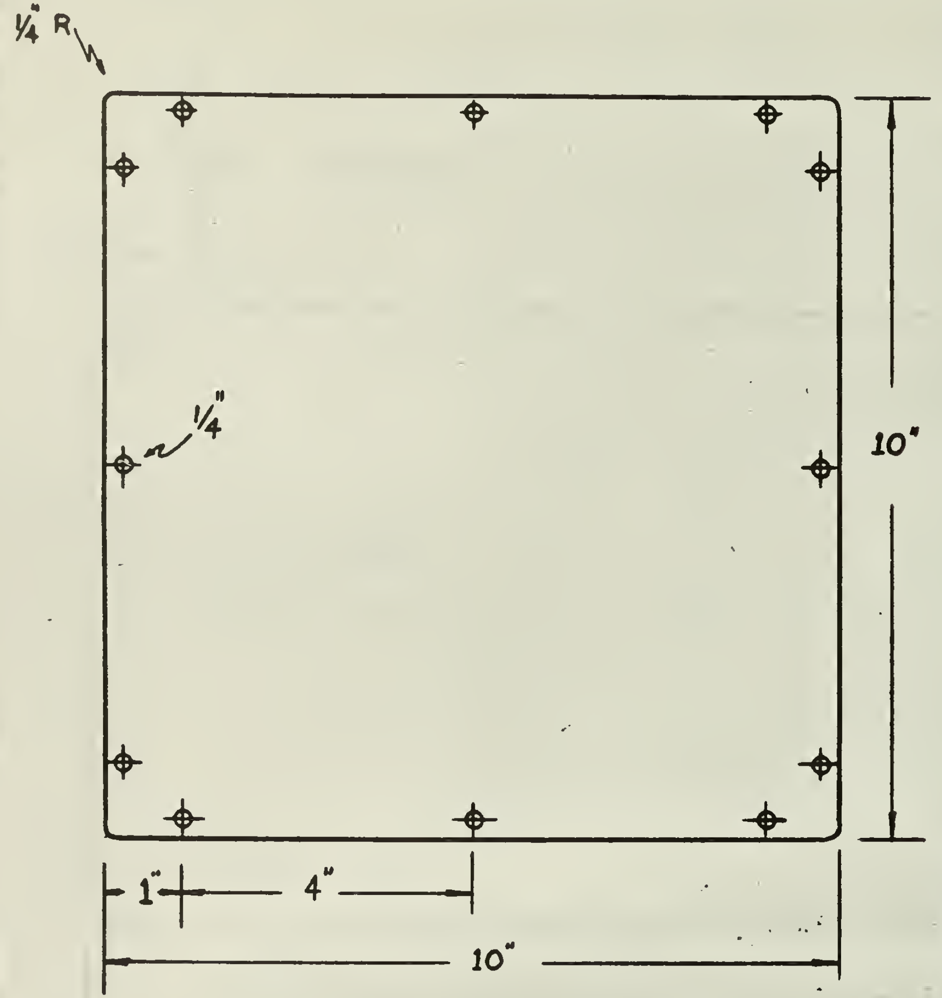
DWG. NO. T602
TANK END
MATERIAL: 1/16" STAINLESS STEEL
SCALE : HALF SIZE
DRAWN BY : Harry J. Howens
28 FEB. 1966



1/4" R
1/4"
10"
4"
1"
1"
4"
23"
I MAKE TWO
U. S. NAVAL POSTGRADUATE SCHOOL
MOTEREY, CALIF.
DWG. NO. T603
TOP, OIL TANK
MATERIAL : 1/16" STAINLESS STEEL
SCALE : ONE THIRD
DRAWN BY :
1 MARCH 1966

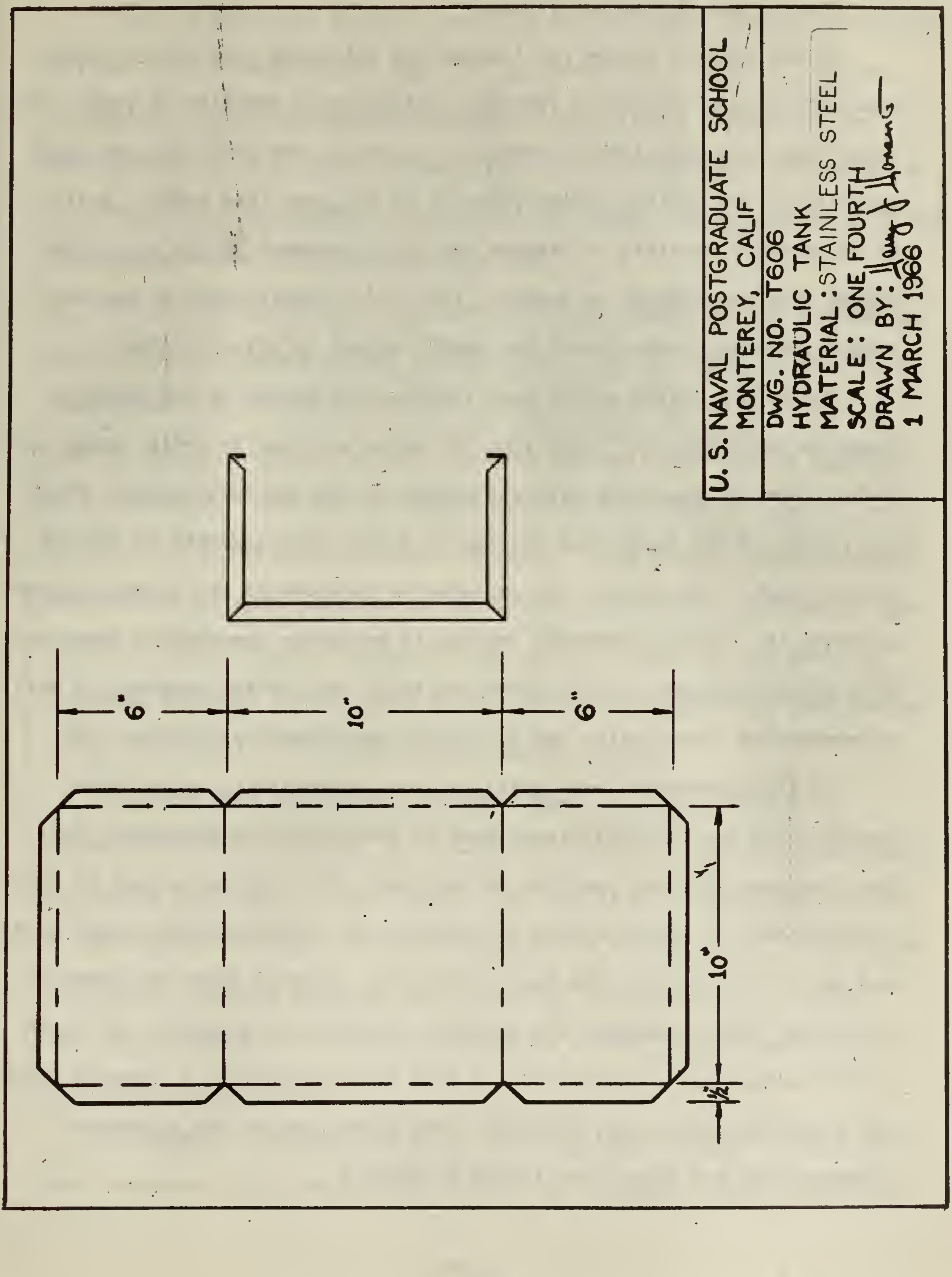


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. T6C4
BODY, OIL TANK
MATERIAL : 1/16" STAINLESS STEEL
SCALE : ONE FOURTH
DRAWN BY :
1 MARCH 1966
6"
10"
6"
23"
1/2
1 TWO REQUIRED



U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. T605
HYDRAULIC TANK TOP
MATERIAL: STAINLESS STEEL
SCALE: HALF
DRAWN BY:
1 MARCH 1966


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF
DWG. NO. T606
HYDRAULIC TANK
MATERIAL : STAINLESS STEEL
SCALE : ONE FOURTH
DRAWN BY:
1 MARCH 1966
6"
10"
6"
10"
1/2

6. Control and Hydraulic System.

In any control scheme for loading and unloading this type of device, some method must be used to indicate position as a function of load. The total load is proportional to both the torque on the shaft and the axial position of the helical pinion relative to the zero load point. While it is entirely possible to measure the axial movement of the pinion and equate this information to loading, the use of strain gages to measure torque provides a more direct and useful method to affect control.

Two ninety degree strain gage rosettes are bonded to the shaft as shown in Table IV. The leads from the gages are lead to radial holes two inches from the gages and into the middle of the test gear shaft. From the center of the shaft they are run to a slip ring assembly on the end of the shaft. The circuit is completed as is shown in the second diagram of Table IV. This arrangement results in automatic temperature compensation and elimination of the effects of axial and bending strains, as well as minimizing inaccuracies due to contact resistance variations. /6/

If left uncovered these gages would be subjected to oil at high temperatures and for this reason must be covered with a protective coating. Because of their position on the shaft, the insulation must be able to withstand the forces caused by rotation, be flexible under torque loads, and act as a barrier to the heat in the oil. To meet these requirements a silicone rubber compound was chosen. In addition to meeting the above listed conditions it resists attack from oils and chemicals, absorbs shock and vibration and is easy to apply. The properties of the particular compound that was chosen are listed in Table V.

Table VI provides loading, position and indicator reading corresponding to load information. A type "N" indicator may be used to indicate loading by adding the tabulated value for the desired loading to the zero reading of the meter. Actual loading is accomplished by regulating the pressure on the piston by adjusting the needle valve controls with the directional valve in the desired position.

The hydraulic system allows for very fine control of the loading but does so with a minimum number of components. An unloading as well as a loading cylinder is needed because although the system will tend to unload by itself, friction is still present and will prevent the complete self unloading. This two way motion could also be used to load in the opposite direction with the motor reversed. This mode of operation would necessitate the zero position of the helical pinion being shifted to the center of the drive gear instead of on one end. This two way mode of operation provides another good reason for strain gages to be employed instead of a position indicator in the control scheme.

The pump unit chosen has a maximum capacity of one tenth of a gallon of fluid per minute, and a maximum pressure of eight hundred pounds per square inch which is well within the safe load carrying capacity of the tubing and fittings. The low flow rate is desirable for this type of loading and the high pressure will allow for any conceivable loadings that may be attempted. The characteristics of the system components are listed in Table VII.

In using the control system and in planning the design, strain readings and pinion displacement information are required. The calculations

that follow are for the design load using a 0.375 inch face width test gear. The following symbols are used in the calculations for this section.

D_i Inner diameter of the test shaft.
$D_i = 0.50$ in.

D_o Outer diameter of test shaft.
$D_o = 2.00$ in.

F Gage factor of the load indicating strain gages.
$F = 2.06$

G Shear modulus.
$G = 12{,}000{,}000$ psi. /15/

J Polar moment of inertia of the test gear shaft.

K Transverse sensitivity factor of the gages.
$K = 0.002$

L_m Axial displacement of pinion from reference.

L_s Distance between helical gear and test gear.
$L_s = 15.0$ in.

R_p Pitch radius of gear.
$R_p = 2.90$ in.

s Arc length displaced.

T Full load torque on the system.
$T = 2900$ pound in.

ψ Helix angle.
$\psi = 12.6$ degrees

ϕ Angle of rotation of one end of the shaft with respect to the opposite end.

γ Shear strain.

The values found in Table VI were the result of a digital computer program using FORTRAN 60 language. The formulas for the printed data are exhibited below. Since the strain gages are mounted on the two inch outside diameter portion of the test gear shaft the first value needed is the polar moment of inertia which is

$$J = \frac{\pi(D_o^4 - D_i^4)}{32} = \frac{(3.14)(2.0IN^4 - 0.5IN^4)}{(32)}$$

$$= 1.565 \text{ in}^4.$$

Then the strain due to design load is

$$\gamma = \frac{TD_o}{2JG} = \frac{(2900 \# \text{ IN})(2 \text{ IN})}{(2)(1.565IN^4)(12 \times 10^6 PSI)}$$

$$= 155 \mu \text{in/in}.$$

If a type "N" strain gage indicator is used as planned, the gage factor, F, will be set in and therefore will not appear in the calculations. A four gage bridge will mean amplified strain readings and thus strain will not be read directly. The reading is also corrected for the transverse sensitivity of the gages themselves however, in this case the correction is negligible. The indicator reading at full load will be /6/

$$\text{Reading} = \frac{4\gamma}{(1 + K)} = \frac{(4)(155)}{(1 + .002)} = 619 \mu\text{volts}.$$

In determining the axial movement of the helical pinion the total angular rotation of the shaft is found to be

$$\phi = \frac{TL_s}{GJ} = \frac{(2900 \# \text{ IN})(15 \text{ IN})}{(12 \times 10^6 \#/IN^2)(1.565 \text{ IN}^4)}$$

$$= 23.2 \times 10^{-4} \text{ radians}.$$

This gives a corresponding arc length of

$$s = 2\phi R_p = (23.2 \times 10^{-4})(2)(2.9 \text{ IN})$$

$$= 136 \times 10^{-4} \text{ in}.$$

Which results in a final theoretical pinion displacement of

$$L_m = \frac{s}{\tan\psi} = \frac{(136 \times 10^{-4} \text{ IN})}{(0.2235)} = 0.0601 \text{ in}.$$

This is a rather small displacement and might tempt one to use a smaller helix angle. If much smaller an angle is chosen the self unloading feature will be lost which is not desirable. Actually because of bearing play and backlash the true displacement will be somewhere between an eighth and a quarter of an inch. /10/

On the following five pages are found the system block diagram, strain gage circuit data, silicone compound characteristics, loading data, and hydraulic systems components list.

HYDRAULIC SYSTEM BLOCK DIAGRAM

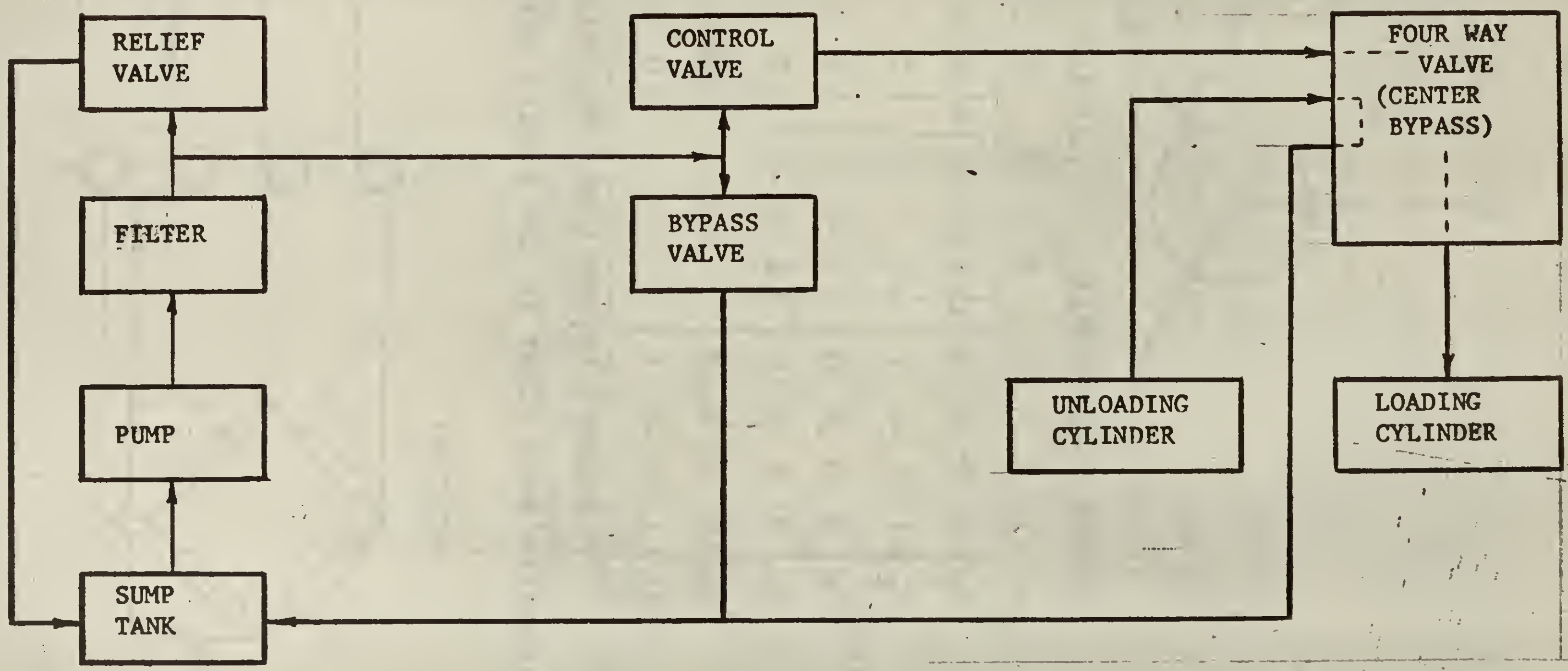
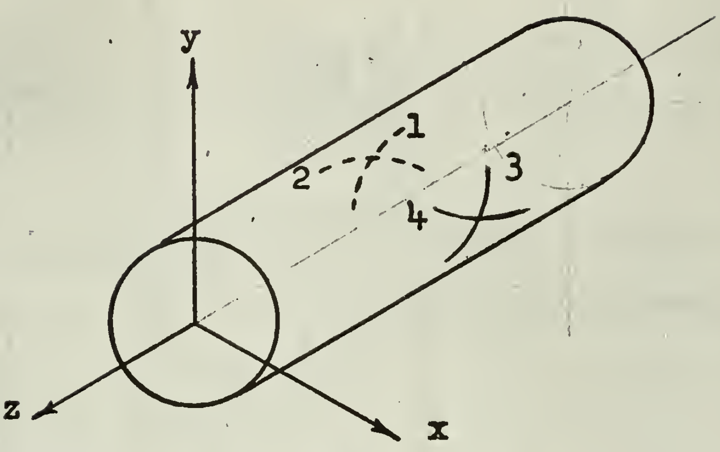


TABLE IV

STRAIN GAGE CIRCUIT DATA



STRAIN GAGE MATRIX $\mu = 0.3$

$$
\begin{array}{c}
 \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z
\end{array}
\begin{array}{c}
\begin{array}{cccc} 1 & 2 & 3 & 4 \end{array} \\
\left[\begin{array}{cccc}
0 & 0 & 0 & 0 \\
-1 & 1 & -1 & 1 \\
\mu & \mu & \mu & \mu \\
0 & 0 & 0 & 0 \\
k & k & k & k \\
1 & -1 & -1 & 1
\end{array}\right]
\end{array}
\times
\begin{array}{c}
\begin{array}{ccc} 2 & 3 & 4 \end{array} \\
\left[\begin{array}{ccc}
-1 & -1 & -1 \\
-1 & 1 & 1 \\
1 & -1 & 1 \\
1 & 1 & -1
\end{array}\right]
\end{array}
=
\begin{array}{c}
\begin{array}{ccc} 2 & 3 & 4 \end{array} \\
\left[\begin{array}{ccc}
0 & 0 & 0 \\
0 & 4 & 0 \\
0 & 0 & 0 \\
0 & 0 & 0 \\
-4\mu & 0 & 0 \\
0 & 0 & -4
\end{array}\right]
\end{array}
$$

To measure torque about the z axis gages 1 and 4 are opposite each other in the strain gage bridge.

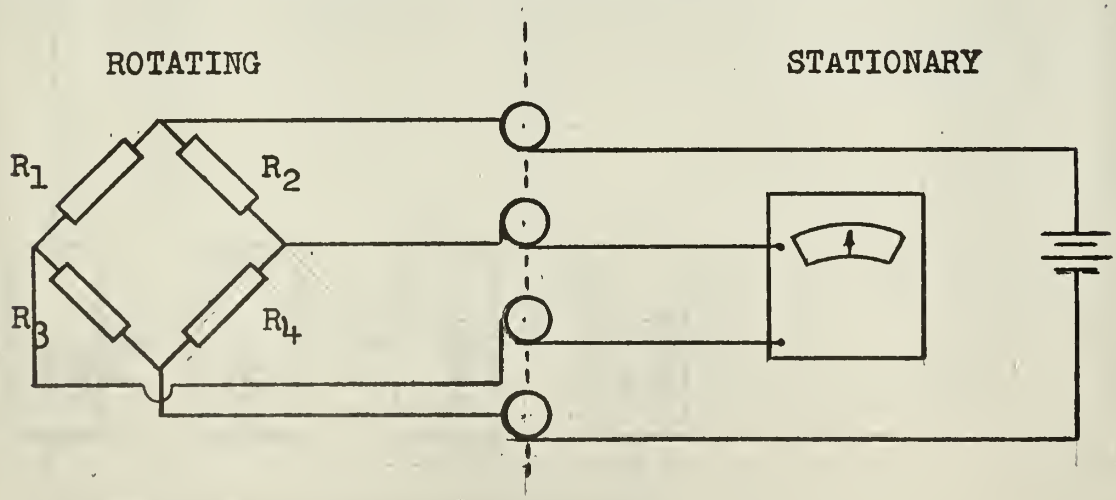


TABLE V

CHARACTERISTICS OF SILICONE COMPOUND

COLOR	RED
SPECIFIC GRAVITY	1.07
HARDNESS SHORE A DUROMETER	33
TENSILE STRENGTH	350 PSI
ELONGATION	400 %
BRITTLE POINT	-90 °F
LINEAR SHRINKAGE	1%
THERMAL CONDUCTIVITY	.12 BTU-FT/HR FT^2°F
COEFFICIENT OF THERMAL EXPANSION	15×10^{-5} IN/IN
DIELECTRIC STRENGTH	500 VOLTS/MIL
DIELECTRIC CONSTANT @ 60 CPS	2.8
DISSIPATION FACTOR @ 60 CPS	.0026
VOLUME RESISTIVITY	3×10^{15} OHM-CM
MAXIMUM TEMPERATURE	600 °F

TABLE VI

LOADING AND METER DATA

NORMAL FORCE POUNDS	LOADING LB/IN	METER READING μVOLTS
20.00	53.33	11.59
40.00	106.67	23.17
60.00	160.00	34.76
80.00	213.33	46.34
100.00	266.67	57.93
120.00	320.00	69.51
140.00	373.33	81.10
160.00	426.67	92.68
180.00	480.00	104.27
200.00	533.33	115.85
220.00	586.67	127.44
240.00	640.00	139.02
260.00	693.33	150.61
280.00	746.67	162.19
300.00	800.00	173.78
320.00	853.33	185.37
340.00	906.67	196.95
360.00	960.00	208.54
380.00	1013.33	220.12
400.00	1066.67	231.71
420.00	1120.00	243.29
440.00	1173.33	254.88
460.00	1226.67	266.46
480.00	1280.00	278.05
500.00	1333.33	289.63
520.00	1386.67	301.22
540.00	1440.00	312.80
560.00	1493.33	324.39
580.00	1546.67	335.98
600.00	1600.00	347.56
620.00	1653.33	359.15
640.00	1706.67	370.73
660.00	1760.00	382.32
680.00	1813.33	393.90
700.00	1866.67	405.49
720.00	1920.00	417.07
740.00	1973.33	428.66
760.00	2026.67	440.24
780.00	2080.00	451.83
800.00	2133.33	463.41
820.00	2186.67	475.00
840.00	2240.00	486.58
860.00	2293.33	498.17
880.00	2346.67	509.76
900.00	2400.00	521.34
920.00	2453.33	532.93
940.00	2506.67	544.51
960.00	2560.00	556.10
980.00	2613.33	567.68
1000.00	2666.67	579.27

TABLE VII

HYDRAULIC SYSTEM COMPONENTS

SUMP CAPACITY	-------------	3 GALLONS
PUMP CAPACITY	-------------	0.1 GAL/MIN, 1725 RPM 0 - 500 POUNDS/INCH^2
MOTOR CHARACTERISTICS	-------------	1/6 HP/1725 RPM 115 VOLT/60 CYCLE
RELIEF VALVE PRESSURE	-------------	360 POUNDS/INCH^2
CONTROL VALVE	-------------	NEEDLE TYPE, 20 TURNS FULL OPEN - FULL CLOSE
BYPASS VALVE	-------------	NEEDLE TYPE, 20 TURNS FULL OPEN - FULL CLOSE
SELECTOR VALVE	-------------	FOUR WAY, CENTER BYPASS
FILTER	-------------	WIRE CLOTH TYPE 25 MICRON

7. Base And Supporting Structure.

The base assembly consists of two eighteen inch "I" beams each ten feet in length, welded to quarter inch steel plates that are bolted to rubber insulated concrete pads. The two beams provide a surface for an upper and lower platform to which the various system components are attached. The lower platform is composed of two sliding sections; this provides for ease of maintenance. The sump tanks and pump-motor combination for the lubrication and hydraulic systems are mounted on this lower platform. Both the lower platform,and upper are made of one inch aluminum plate. It serves as a base for the test machine, drive motor, instrument panel and the two heat exchangers. These last items are mounted on the underside of the table directly below the oil pan drains.

The base plate, B204, is bolted and pinned directly to the table; it is fabricated from one inch steel plate. The bearing hangers, B202 and B203, are attached to the base plate with pins and screws. At first glance the hangers appear oversize but because of the large twist inherent in using the four square principle their beefiness is highly desirable. Besides resisting twisting they also serve as an anchor and support for the hydraulic assembly and are a housing for the bearing adjusters, B201. Rather than mount the bearings in a rigid frame it was decided that a means of compensating for expansion and load effects should be provided. Aside from the obvious benefits of adjustment during operation, the adjusters allowed for a relaxation of the tolerances on the shafts, base plate and bearing hangers. In the part drawings the hangers appear as one piece, however they are a two piece assembly. The

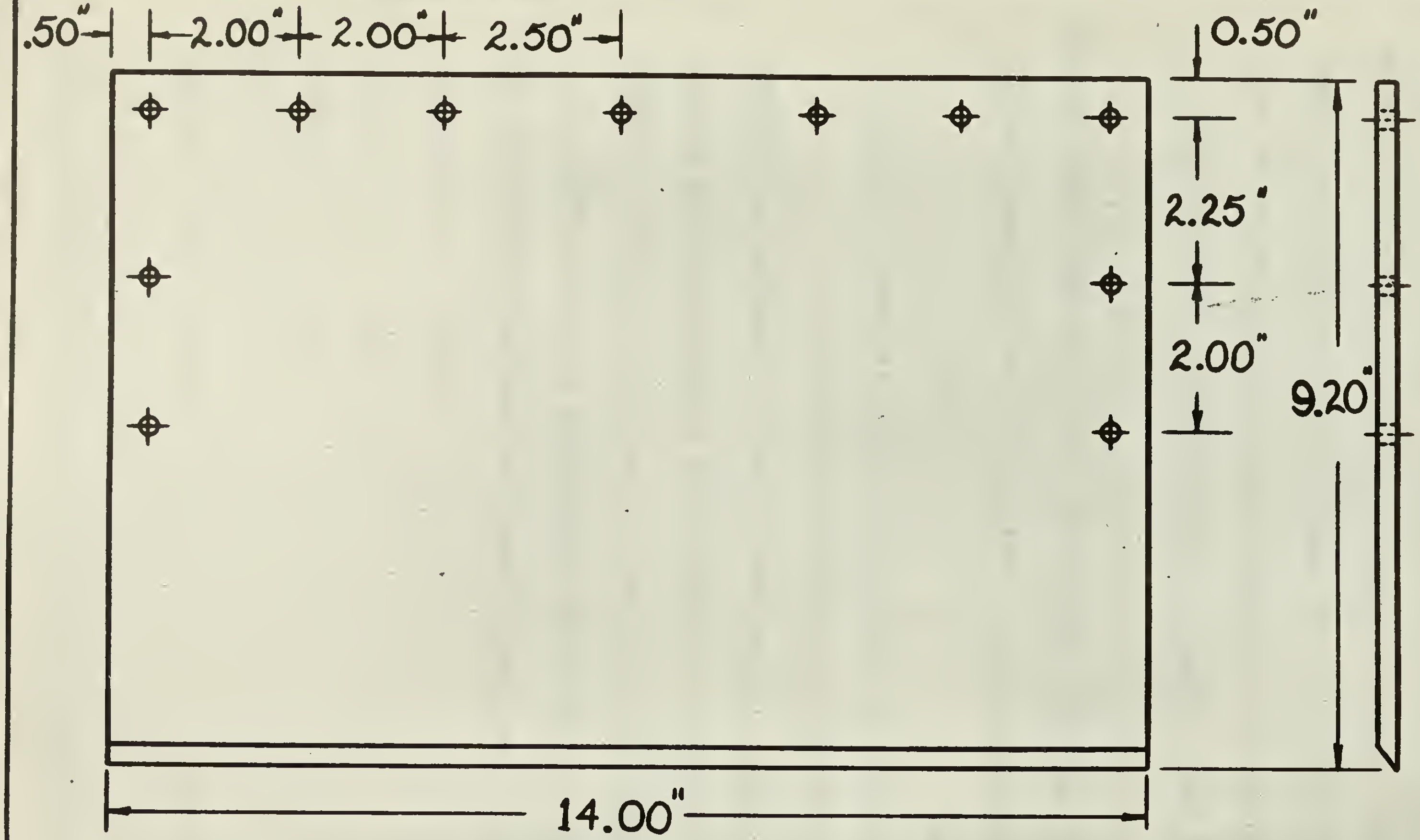
base and main section were pinned and screwed together. This was done to save on material and simplify fabrication. The recessed grooves near the base are for the oil pans, L301 and L302. To ensure a complete seal silicone rubber splash flaps are installed above, and extending into the pans. They are held in place by the same silicone compound used before to protect the strain gages. To ensure complete isolation of the two sections a 1/32 inch thick silicone rubber gasket is employed at all metal junctions.

The sides, B206 and B207, and the top, B208, are fabricated from one quarter inch stainless steel plate. This was chosen for both strength and cosmetic reasons.

The combination view and access port, B209, is made of pyrex which will withstand the splash effects of the hot oil.

A two by three foot aluminum control panel is mounted on brackets that are screwed to the upper platform. The brackets are made up from angle iron using welded construction. All switches, controls, valves and meters are mounted on this panel and marked with tape labels.

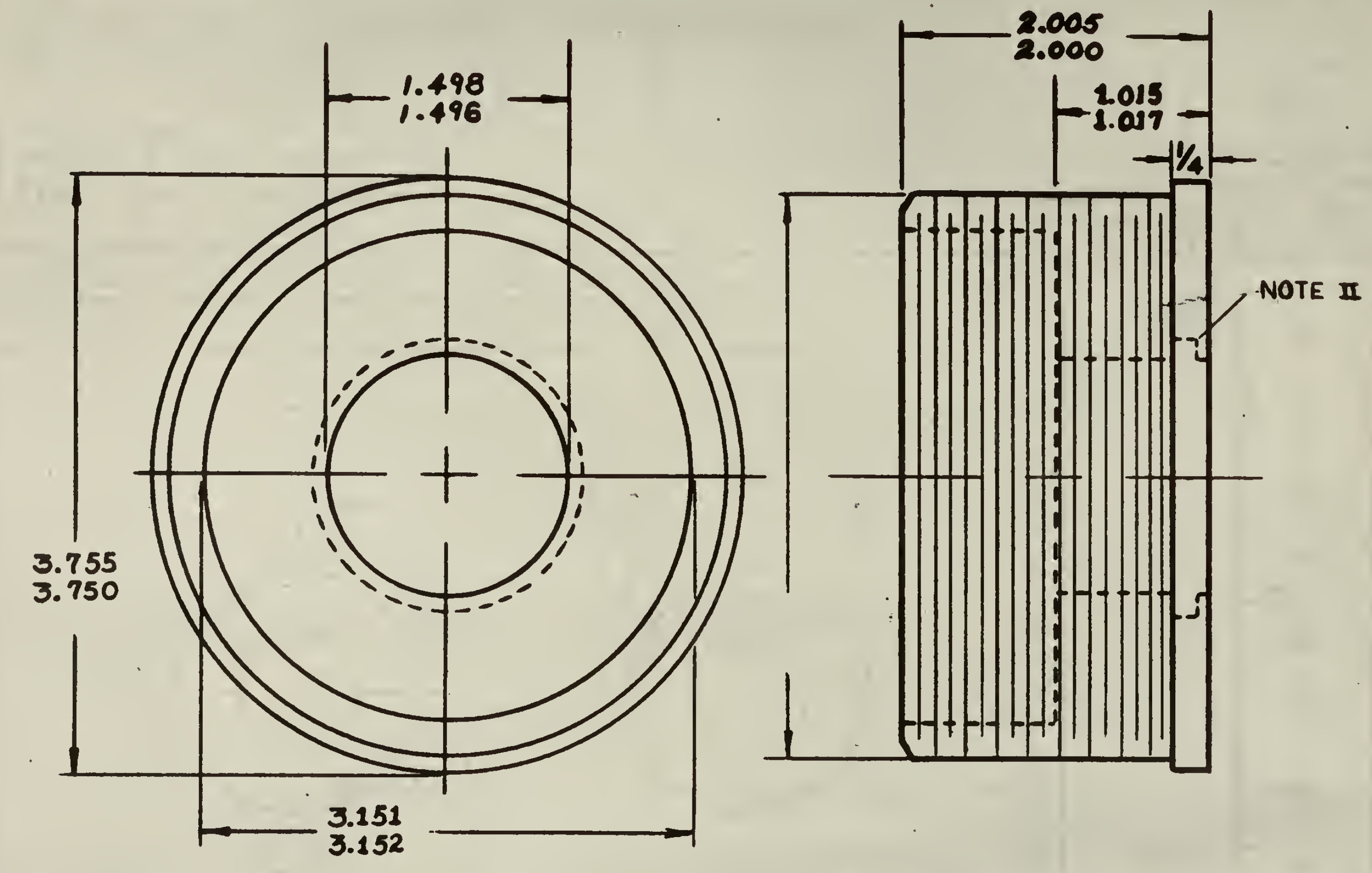
The following thirteen pages contain the various drawings for this section.



I ALL HOLES ¼" DIAMETER

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. B 209
VIEW PORT
MATERIAL : PYREX
DRAWN BY:
17 MARCH 1966

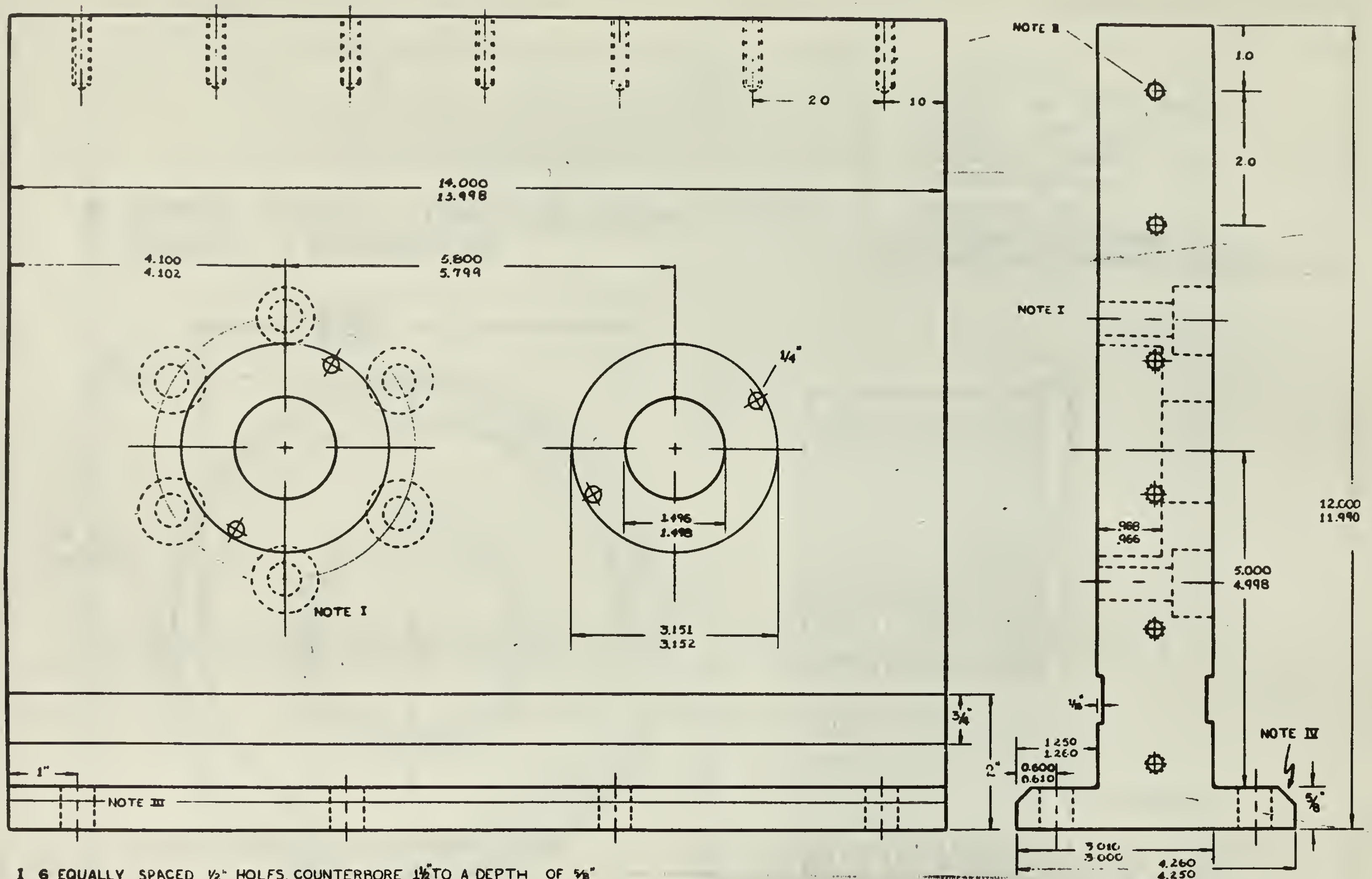


I THREAD 3.5-20UNF-2A

II GROOVE DIAMETER 1.751/1.753, WIDTH .215/.217

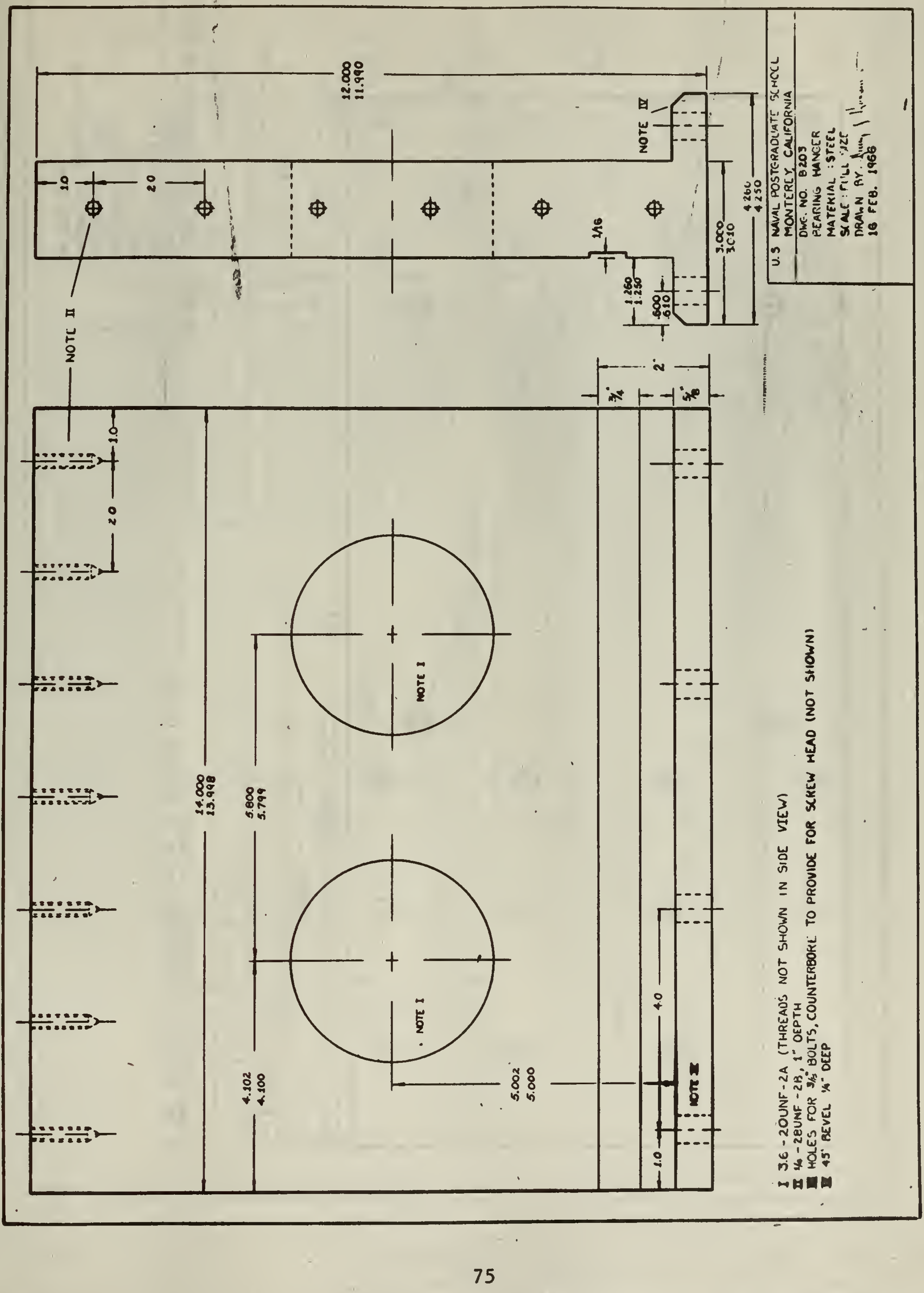
U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA

DWG. NO. B201
BEARING ADJUSTER
MATERIAL: STEEL
SCALE: FULL SIZE
DRAWN BY:
13 FEB. 1968

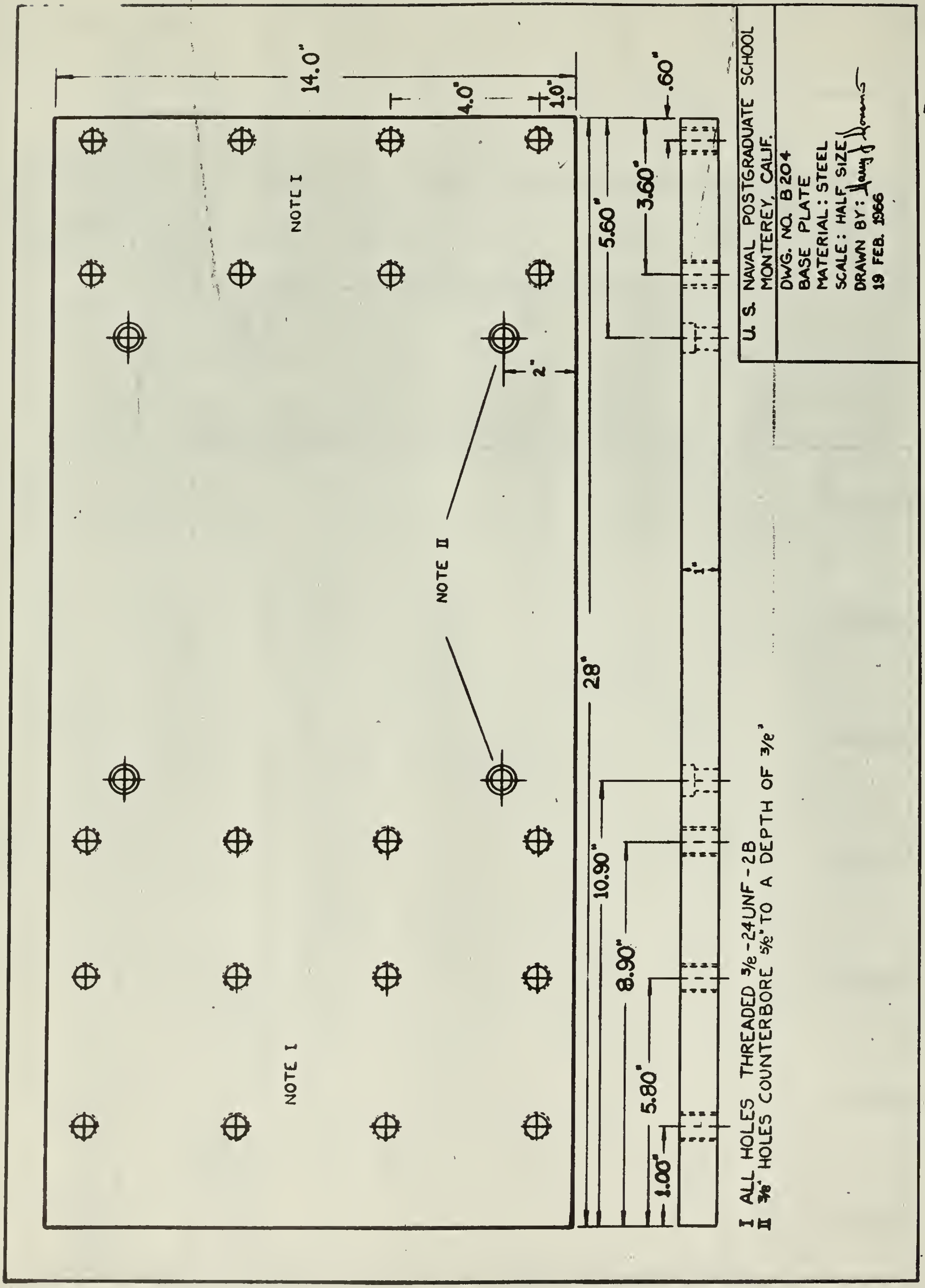


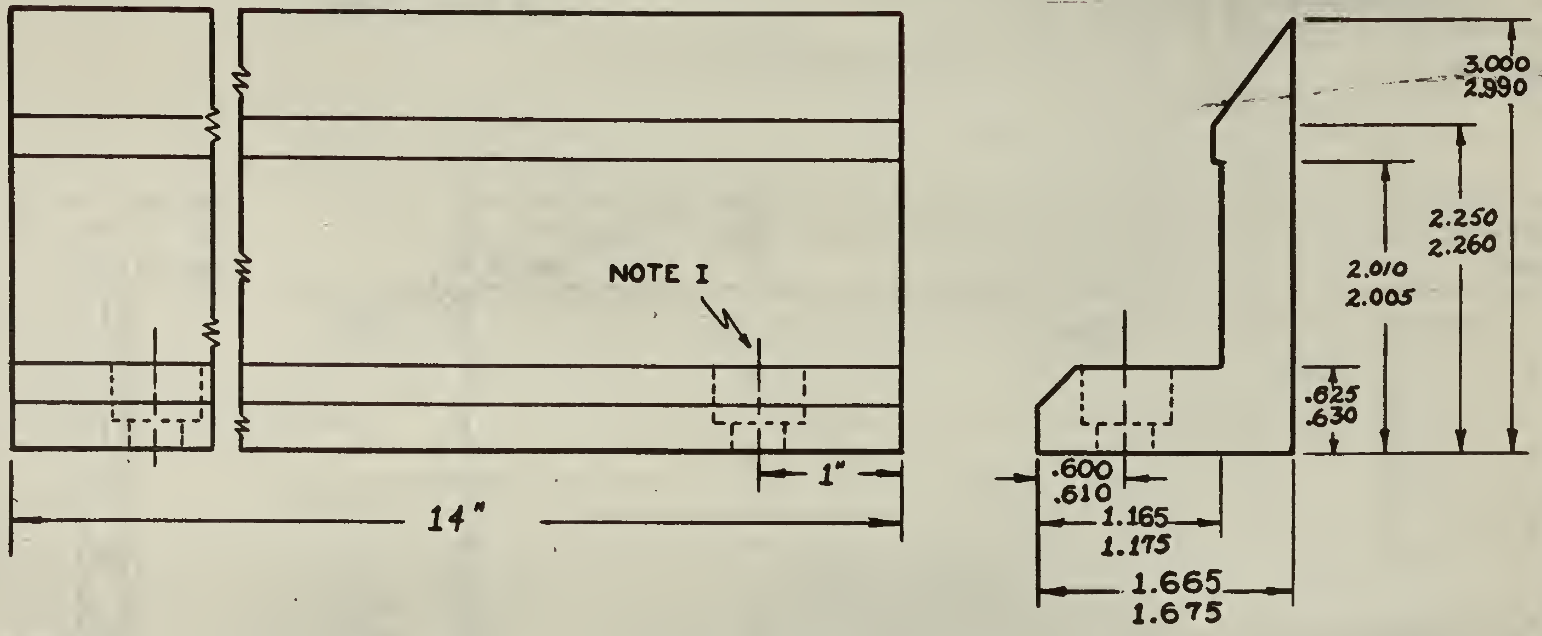
I 6 EQUALLY SPACED 1/2" HOLES, COUNTERBORE 1 1/2" TO A DEPTH OF 5/8"
II 1/4"-28UNF-2B, 1" DEPTH, 2" SPACING OF HOLES
III HOLES FOR 5/8" BOLTS, 4" SPACING OF HOLES
IV 45° BEVEL 1/4" DEEP

U.S. NAVAL POSTGRADUATE SCHOOL
MONEREY, CALIFORNIA
DRW. NO. B 202
BEARING HANGER, TEST END
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY : [signature]
15 FEB 1966



U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
DWG. NO. B203
BEARING HANGER
MATERIAL : STEEL
SCALE : FULL SIZE
DRAWN BY:
16 FEB. 1968
12.000
11.990
NOTE II
NOTE IV
1.0
2.0
1/16
3.000
3.010
4.260
4.250
1.260
1.250
.600
.610
2"
3/4"
5/8"
14.000
13.998
5.800
5.799
NOTE I
4.102
4.100
5.002
5.000
4.0
1.0
NOTE III
I 3.6 -20UNF-2A (THREADS NOT SHOWN IN SIDE VIEW)
II 1/4 -28UNF -2B, 1" DEPTH
III HOLES FOR 3/8" BOLTS, COUNTERBORE TO PROVIDE FOR SCREW HEAD (NOT SHOWN)
IV 45° BEVEL 1/4" DEEP

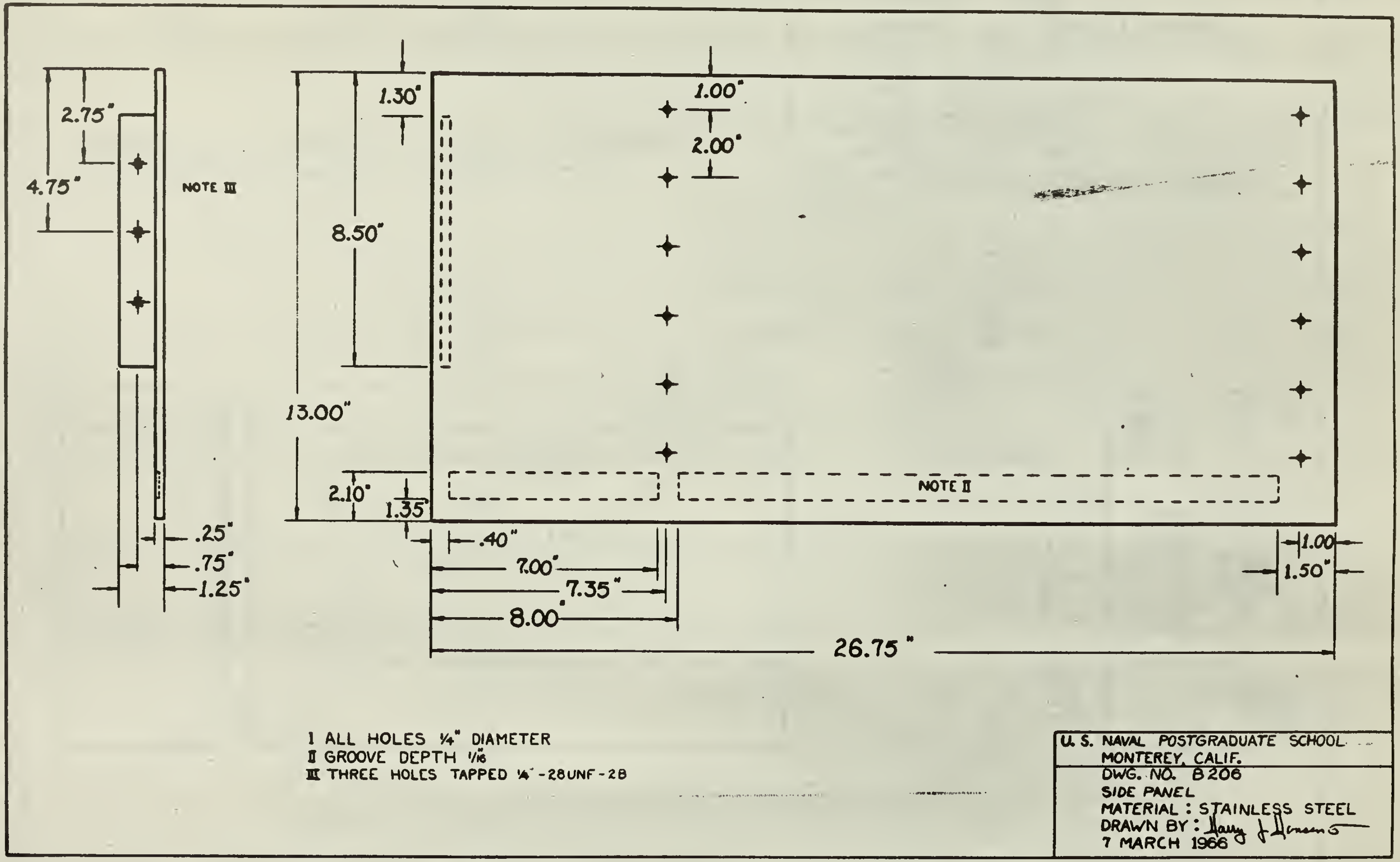

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. B204
BASE PLATE
MATERIAL: STEEL
SCALE: HALF SIZE
DRAWN BY:
19 FEB. 1966
I ALL HOLES THREADED 3/8-24UNF-2B
II 3/8" HOLES COUNTERBORE 5/8" TO A DEPTH OF 3/8"
NOTE I
NOTE II
14.0"
4.0"
1.0"
.60"
5.60"
3.60"
2"
28"
1"
10.90"
8.90"
5.80"
1.00"



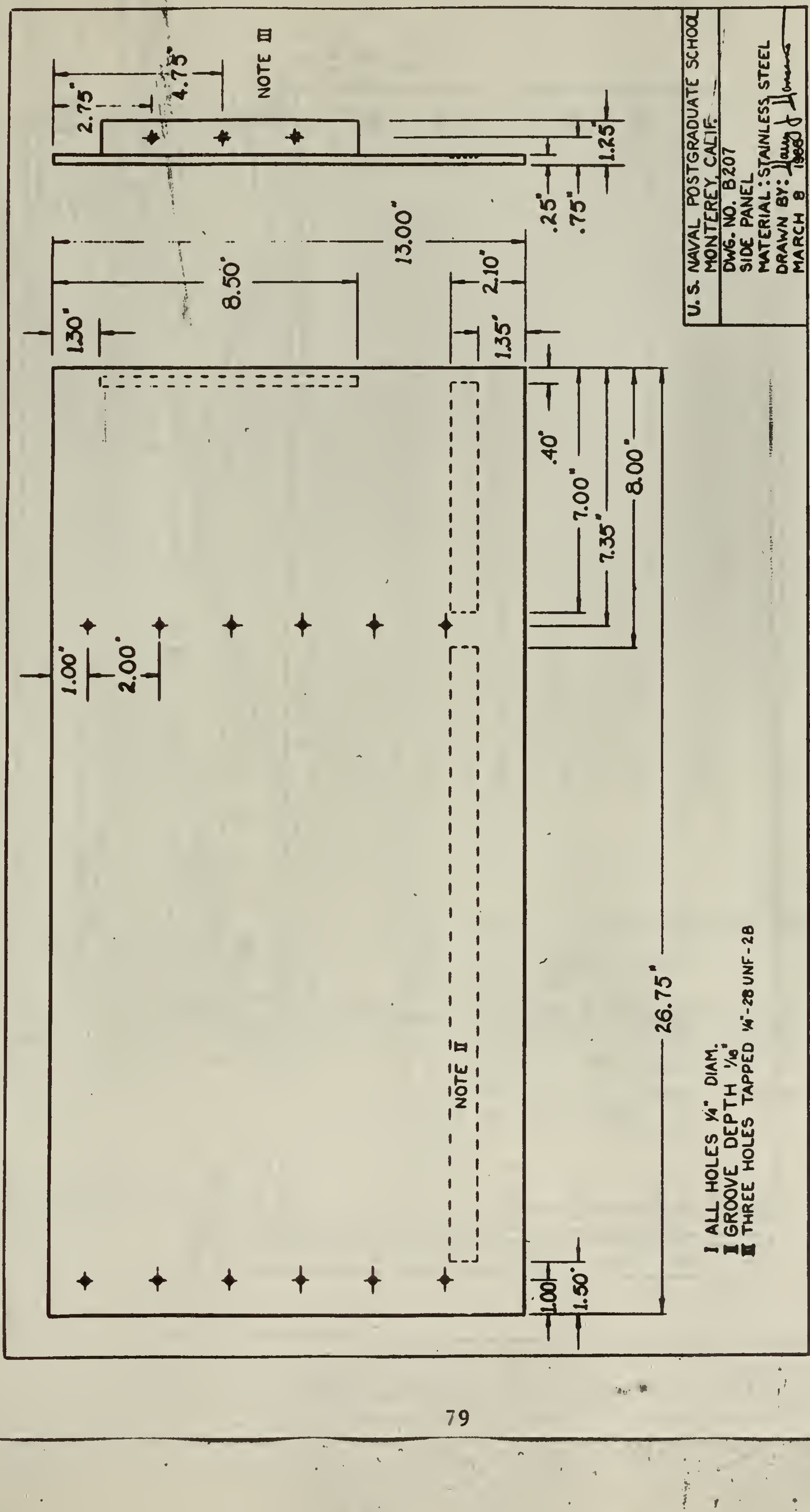
I CLEARANCE HOLES FOR 3/8" CAP SCREWS, 4" SPACING BETWEEN SCREWS

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

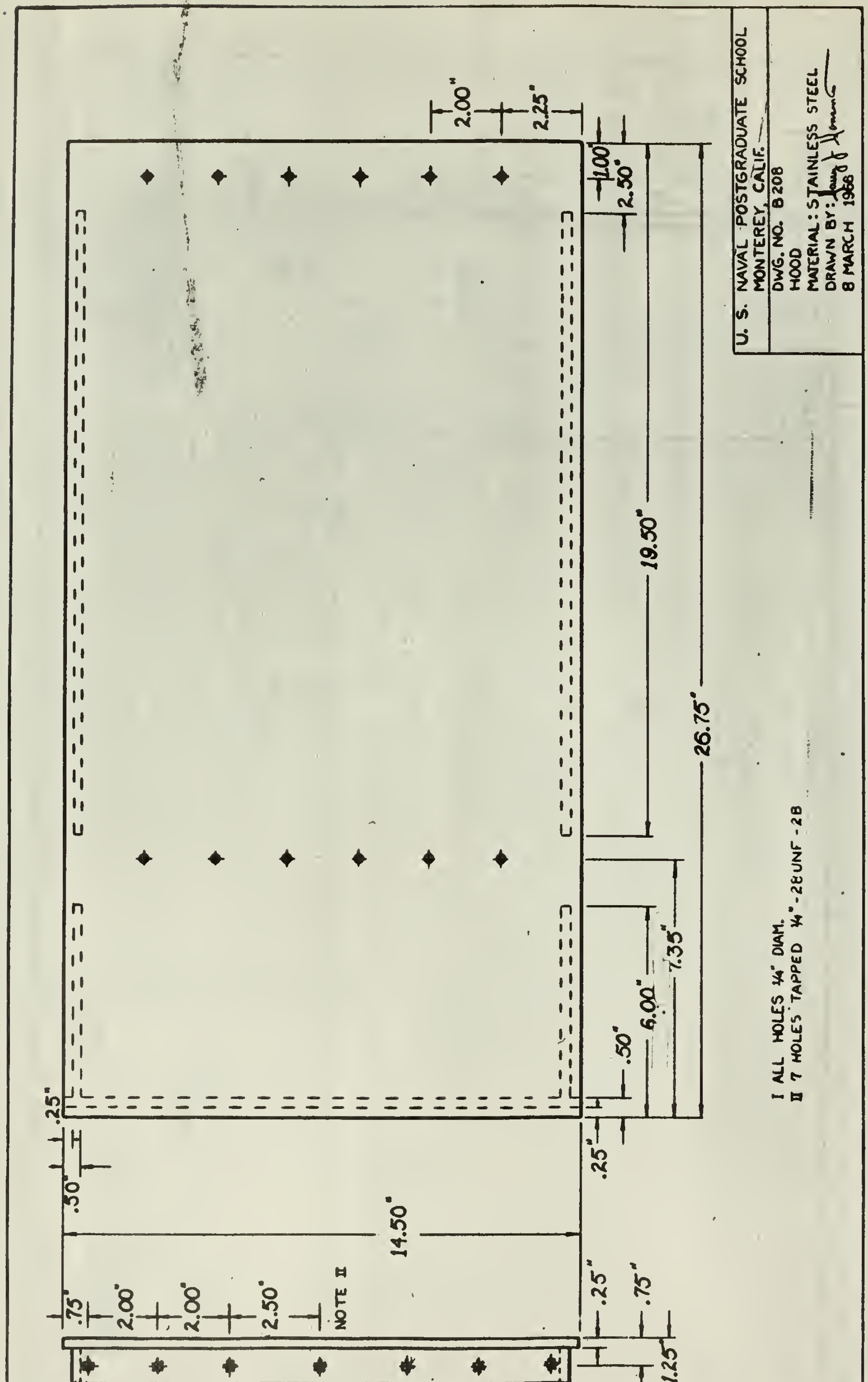
DWG. NO. B 205
END, TEST SECTION
MATERIAL : STEEL
DRAWN BY : Harry J. Hansen
4 MARCH 1966

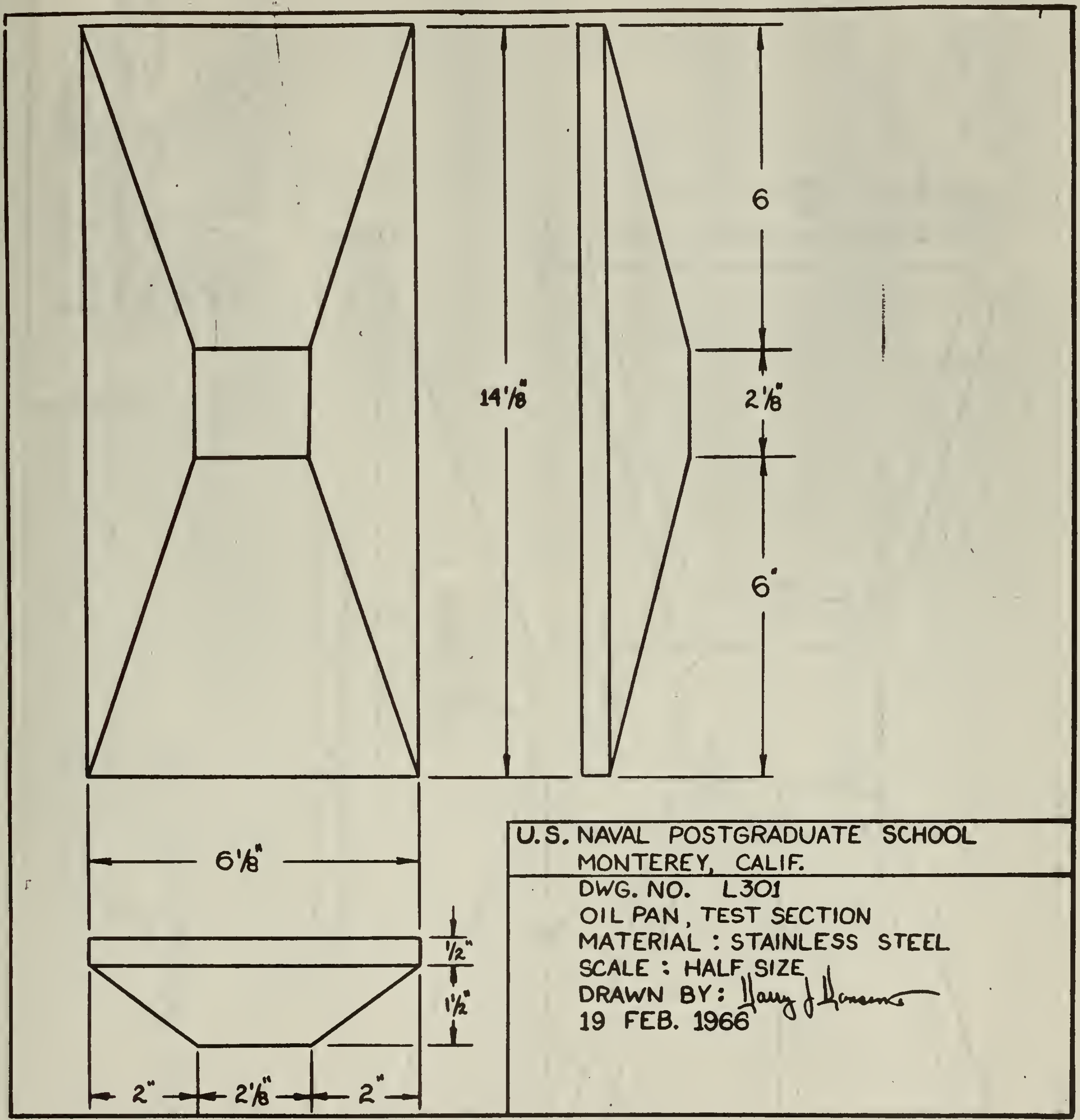


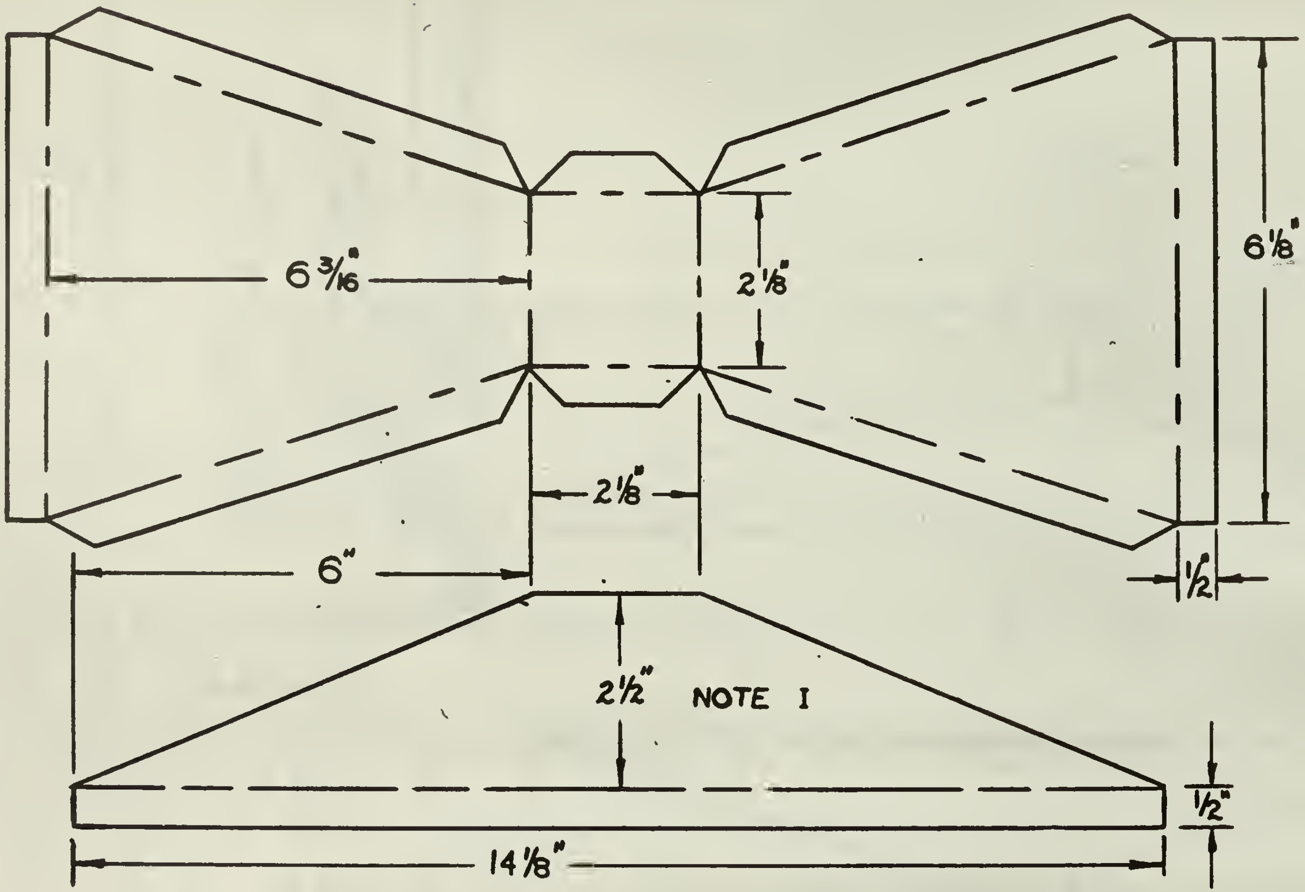
2.75"
4.75"
NOTE III
.25"
.75"
1.25"
1.30"
8.50"
13.00"
2.10"
1.35"
1.00"
2.00"
NOTE II
.40"
7.00"
7.35"
8.00"
26.75"
1.00"
1.50"
I ALL HOLES 1/4" DIAMETER
II GROOVE DEPTH 1/16
III THREE HOLES TAPPED 1/4"-28UNF-2B
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. B206
SIDE PANEL
MATERIAL : STAINLESS STEEL
DRAWN BY :
7 MARCH 1966



U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. B207
SIDE PANEL
MATERIAL: STAINLESS STEEL
DRAWN BY:
MARCH 8 1968
I ALL HOLES 1/4" DIAM.
II GROOVE DEPTH 1/16"
III THREE HOLES TAPPED 1/4"-28 UNF-28
NOTE II
NOTE III
26.75"
13.00"
8.50"
8.00"
7.35"
7.00"
4.75"
2.75"
2.10"
2.00"
1.50"
1.35"
1.30"
1.25"
1.00"
.75"
.40"
.25"


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. B208
HOOD
MATERIAL: STAINLESS STEEL
DRAWN BY:
8 MARCH 1968
I ALL HOLES 1/4" DIAM.
II 7 HOLES TAPPED 1/4"-28UNF-2B
NOTE II
2.00"
2.25"
1.00"
2.50"
19.50"
26.75"
7.35"
6.00"
.50"
.25"
.30"
14.50"
.75"
2.00"
2.00"
2.50"
.25"
.75"
1.25"

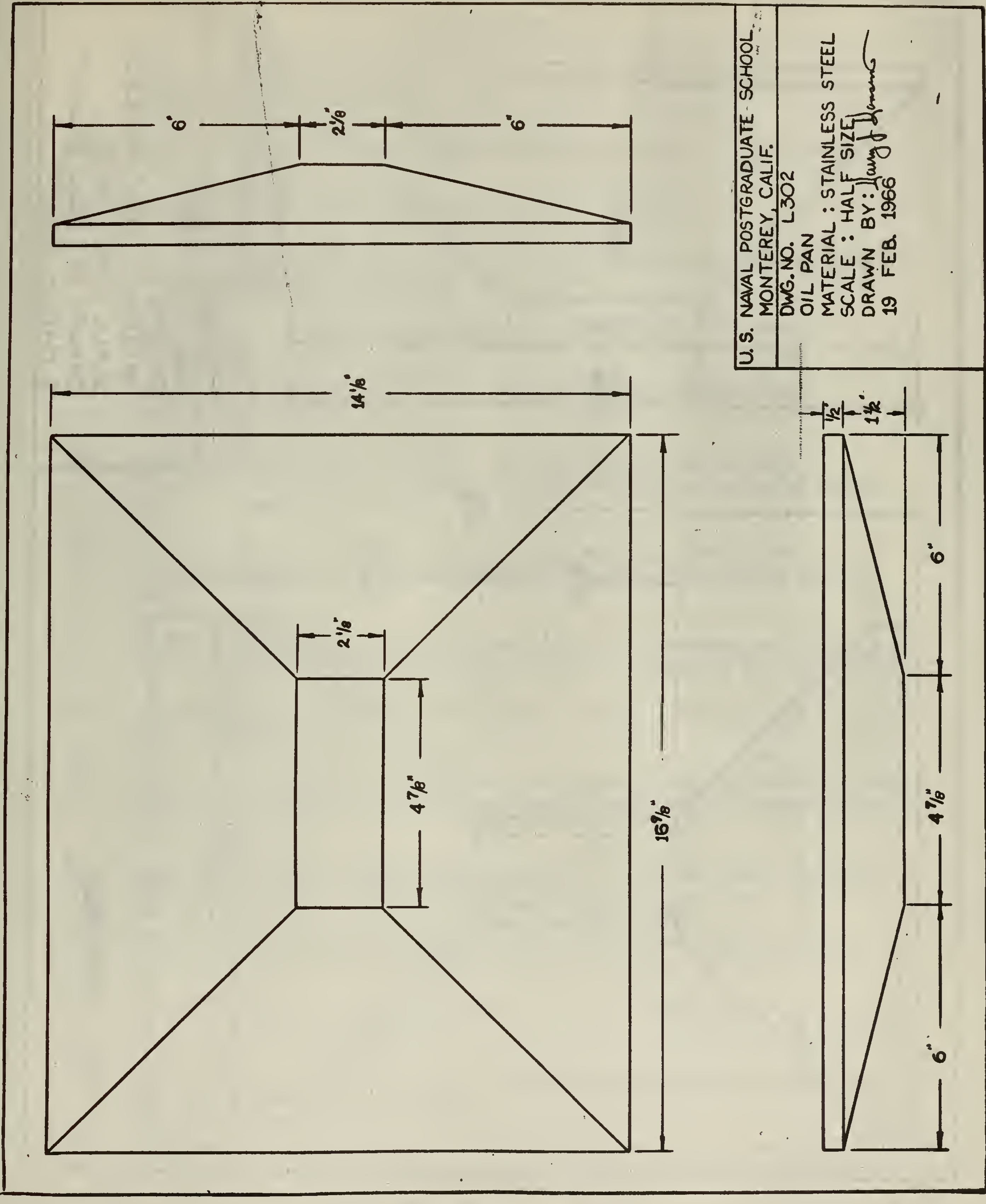

6
14 1/8"
2 1/8"
6"
6 1/8"
1/2"
1 1/2"
2"
2 1/8"
2"
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. L301
OIL PAN, TEST SECTION
MATERIAL : STAINLESS STEEL
SCALE : HALF SIZE
DRAWN BY: Harry J. Hansen
19 FEB. 1966

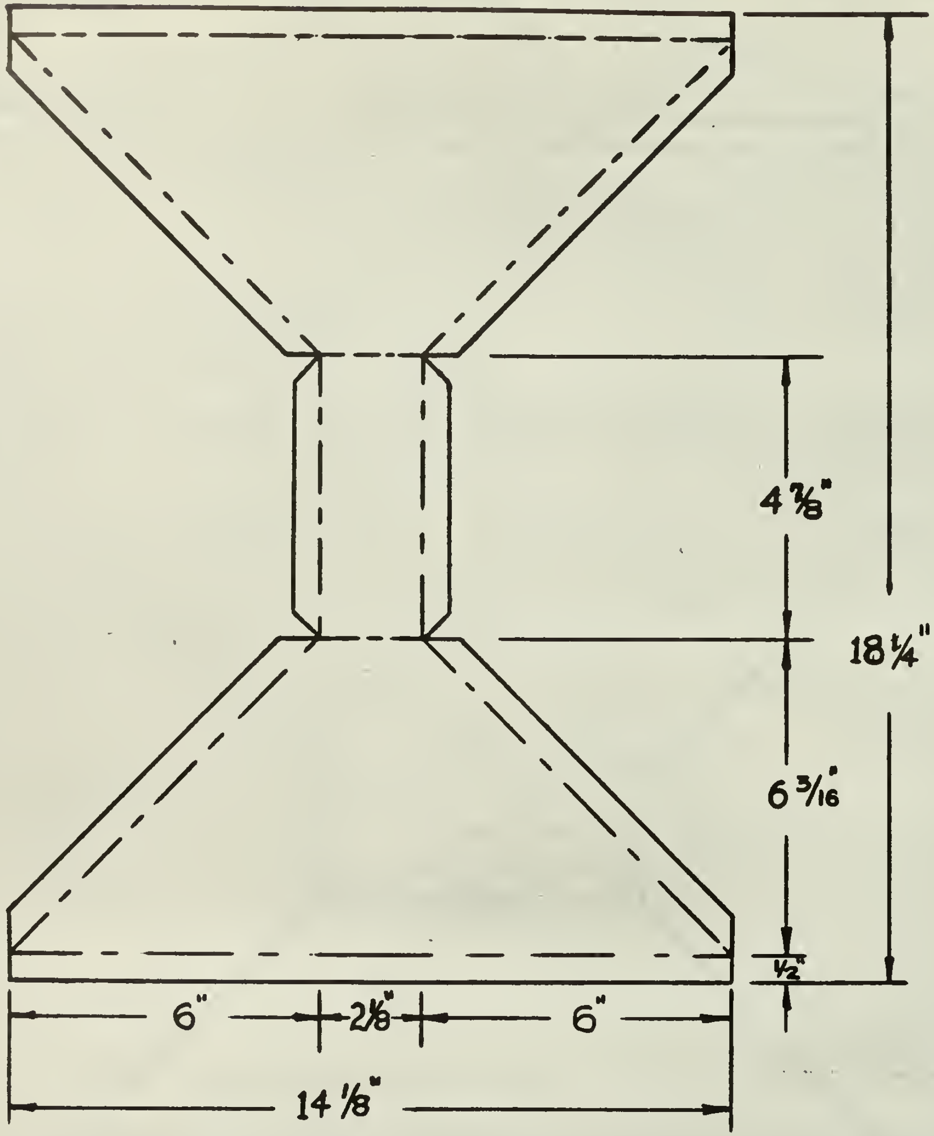


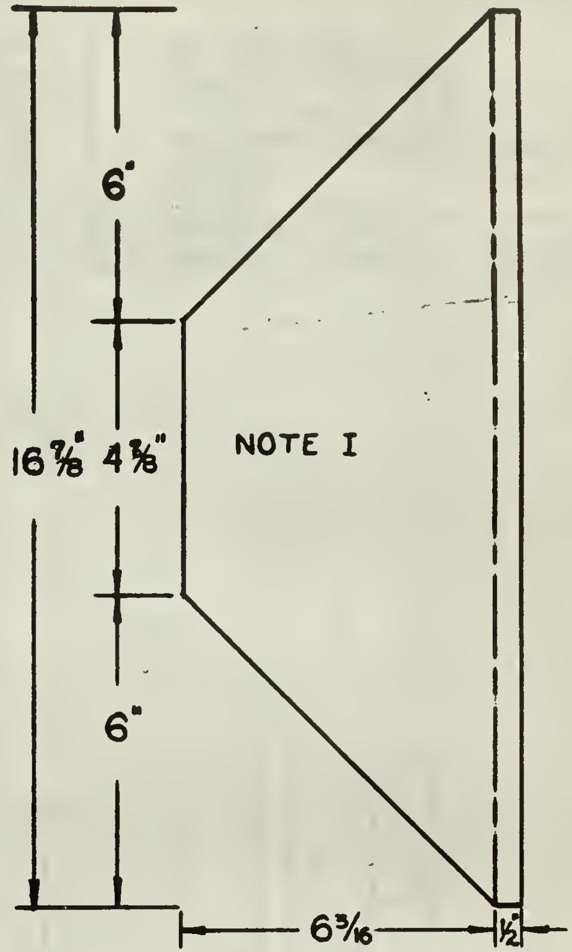
I TWO REQUIRED PER PAN

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. L304
OIL PAN, TEST SECTION
MATERIAL : STAINLESS STEEL
DRAWN BY :
6 MARCH 1966


U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. L302
OIL PAN
MATERIAL : STAINLESS STEEL
SCALE : HALF SIZE
DRAWN BY:
19 FEB. 1966
6"
2 1/8"
6"
14 1/8"
2 1/8"
4 7/8"
16 7/8"
1/2
1 1/2"
6"
4 7/8"
6"





I TWO REQUIRED

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. L303
OIL PAN
MATERIAL : .020" STAINLESS
SCALE : ONE THIRD
DRAWN BY: Harry J. Hansen
2 MARCH 1966

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APPENDIX I

GEAR DATA

The gears used in this machine were not designed by the author and were obtained by contract purchase. The dimensions and heat treatment specifications follow AGMA standards for precision gearing. The basic dimensions and other pertinent data for the four gears used is tabulated below.

	DRIVE GEAR G705	TEST GEAR G704	HELICAL DRIVE G701	HELICAL PINION G702
FACE WIDTH	0.5625	0.375	4.000	2.000
PITCH DIAMETER	5.7795	5.7795	5.800	5.800
NUMBER OF TEETH	65	65	68	68
NORMAL PRESSURE ANGLE	20°	20°	20°	20°
HELIX ANGLE	-	-	12°18′33″	12°18′33″
TRANSVERSE PRESSURE ANGLE	-	-	20°25′55″	20°25′55″
TRANSVERSE PITCH	-	-	11.72413	11.72413
BACKLASH WITH MATE	.003/.005	.003/.005	.003/.005	.003/.005
OUTSIDE DIAMETER	5.9877 5.9847	5.9877 5.9847	5.9667 5.9617	5.9667 5.9617
BASE CIRCLE DIAMETER	5.4309	5.4309	5.4351	5.4351
NORMAL CIRCULAR TOOTH THICKNESS	0.1440 0.1425	0.1440 0.1425	0.1294 0.1284	0.1294 0.1284
WHOLE DEPTH	0.2083	0.2083	0.1958	0.1958

As a check on the maximum stress on the gear teeth the Lewis equation is solved. The symbols defined below are used in the calculations.

b — Face width of the test gear.
$b = 0.375$ in.

F_d — Maximum dynamic load.
$F_d = 1880$ pounds.

K_f — Strength reduction factor.
$K_f = 1.4$ /2/

P_d — Diametral pitch of test gear.
$P_d = 12$

s — Maximum stress on gear tooth.

s_u — Ultimate stress.
$s_u = 174$ ksi for 9310 steel. /2/

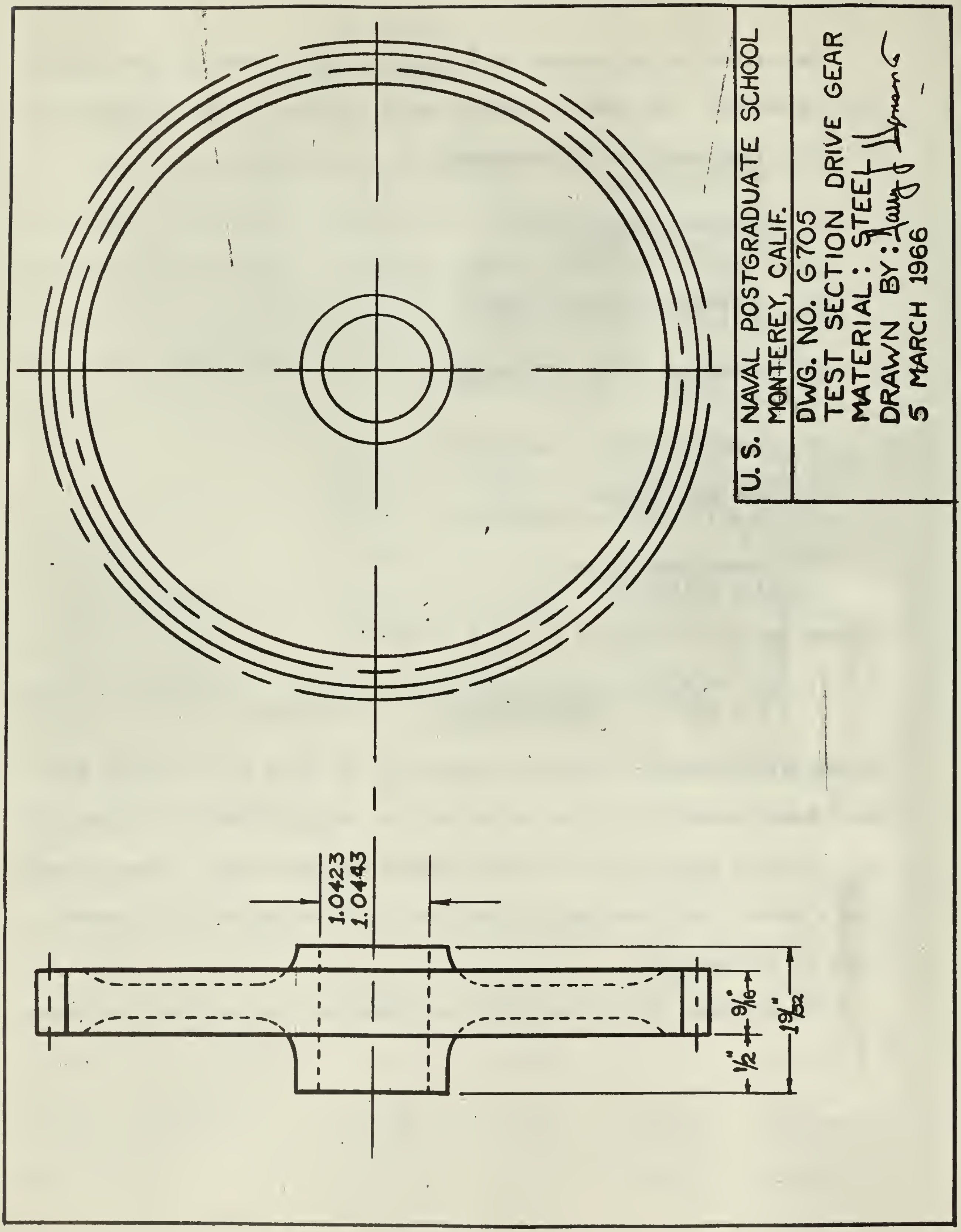
Y — Lewis form factor.
$Y = 0.721$ /2/

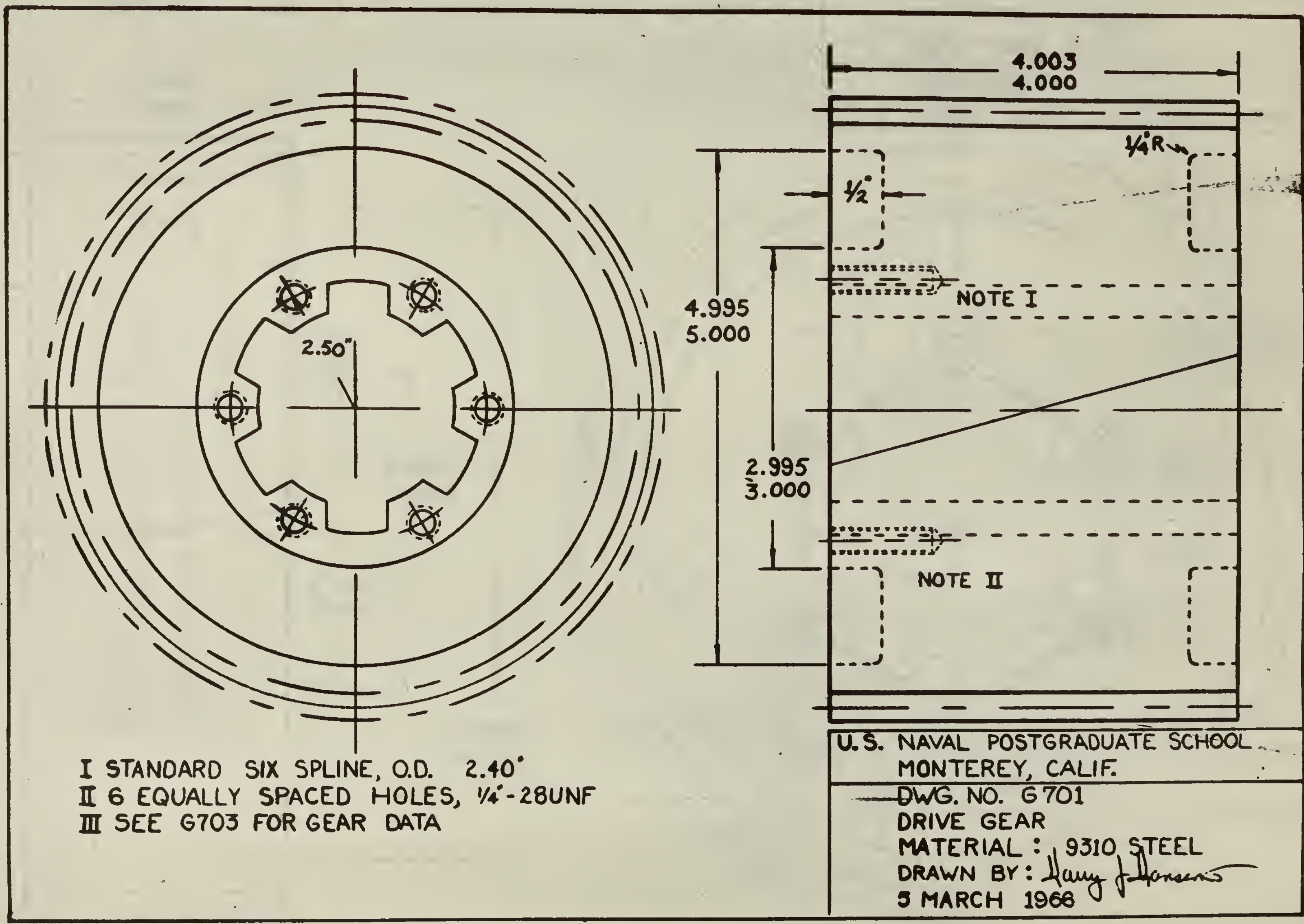
Solving the Lewis equation yields a stress of

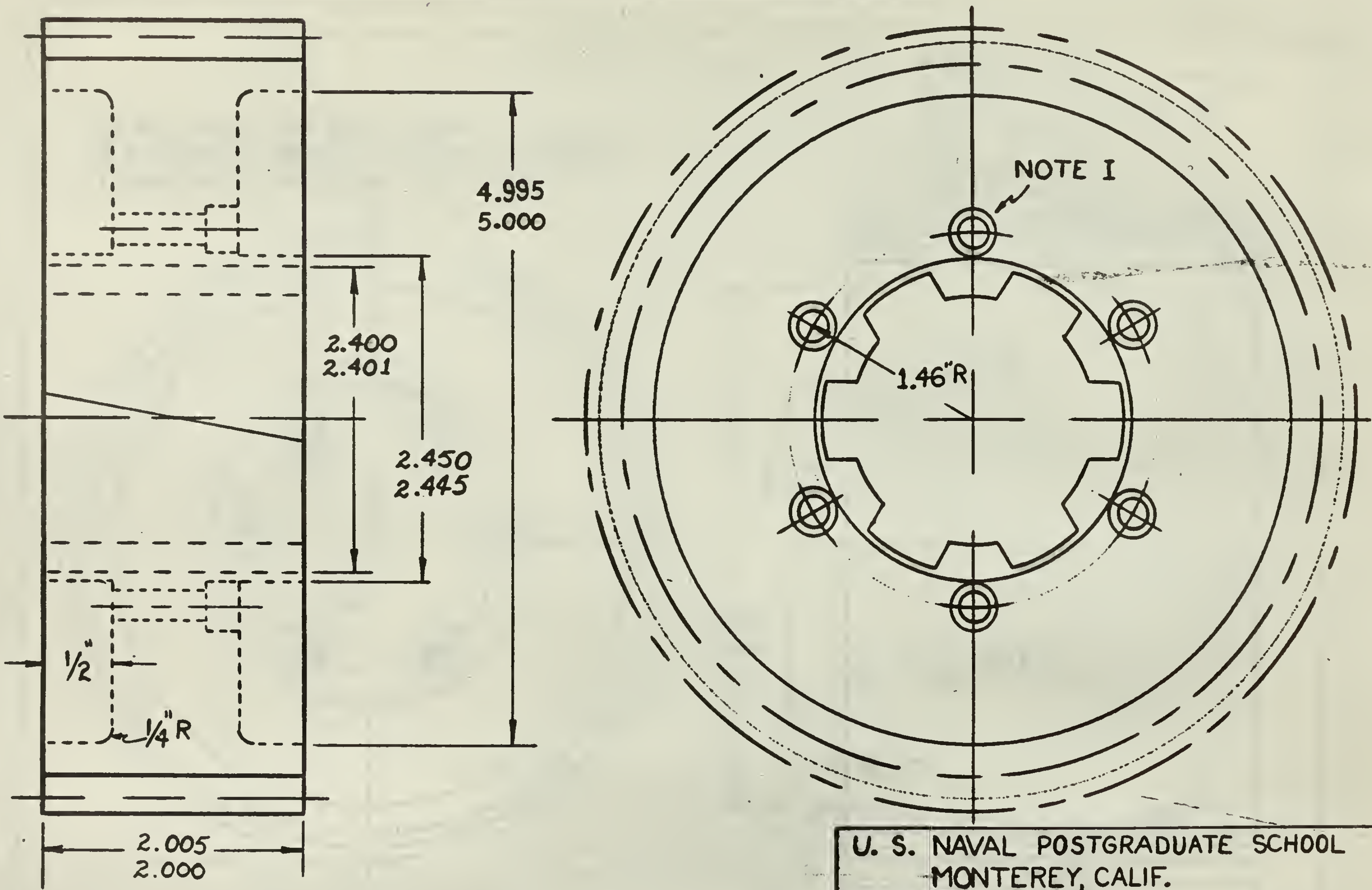
$$s = \frac{F_d P_d K_f}{bY} = \frac{(1880)(12)(1.4)}{(0.375)(0.721)} = 117{,}000 \text{ psi.}$$

Faires states that the fatigue strength may be taken as $60 + 0.2s_u$ ksi which would result in a value of 95 ksi for the gear material. From this it would appear that at higher speeds and thus higher dynamic loads that a wider faced test gear be used and thus stay below the endurance limit of the material.

The drawings of the four gears are presented on the next four pages.


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. G705
TEST SECTION DRIVE GEAR
MATERIAL: STEEL
DRAWN BY:
5 MARCH 1966
1.0423
1.0443
1/2"
9/16"
1 9/32"


4.003
4.000
1/4" R
1/2"
NOTE I
NOTE II
4.995
5.000
2.995
3.000
2.50"
I STANDARD SIX SPLINE, O.D. 2.40"
II 6 EQUALLY SPACED HOLES, 1/4"-28UNF
III SEE G703 FOR GEAR DATA
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. G701
DRIVE GEAR
MATERIAL : 9310 STEEL
DRAWN BY: Harry J. Hansen
5 MARCH 1968

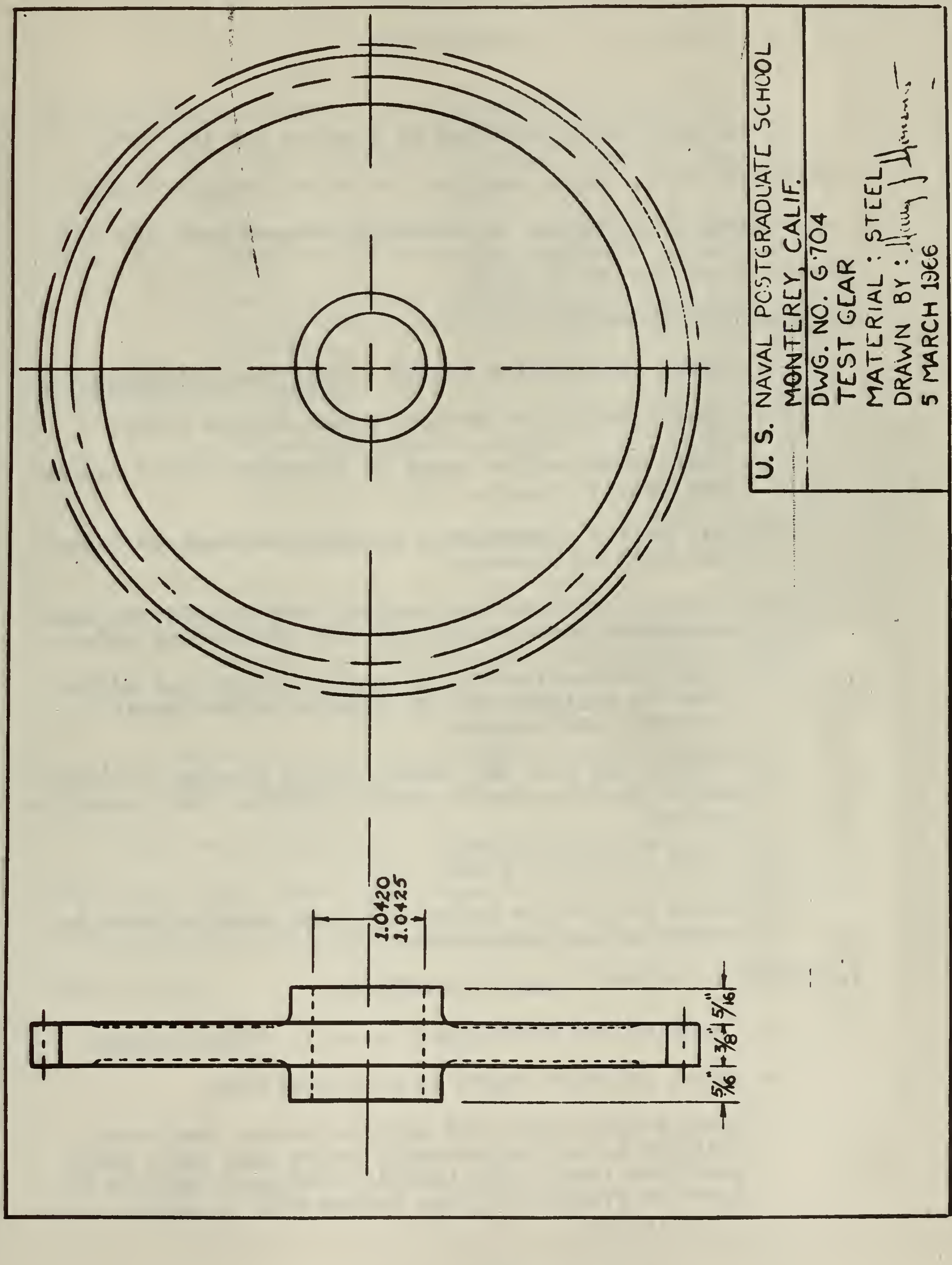


I 6 EQUALLY SPACED CLEARANCE HOLES FOR 1/4" CAP SCREWS

II SEE G703 FOR GEAR DATA

U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.

DWG. NO. G702
HELICAL PINION
MATERIAL: 9310 STEEL
DRAWN BY: Harry J. Hanson
3 MARCH 1966


U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIF.
DWG. NO. G704
TEST GEAR
MATERIAL : STEEL
DRAWN BY :
5 MARCH 1966
1.0420
1.0425
5/16"
3/8"
5/16"

APPENDIX II

OPERATION

This section is provided as a guide for the safe operation of the whole machine. It is not intended that any testing proceedures be included because they are too numerous and varied to list.

1. STARTING PROCEDURE::

   a. Close lubrication system supply needle valves.

   b. Open lubrication system bypass valves fully.

   c. Start lubrication pumps to circulate fluid around the bypass loop.

   d. Set heater temperature controls to desired level and turn on heaters.

   e. When oil reaches the desired temperature set heat exchanger thermostats and turn on cooling water.

   f. Open lubrication supply needle valves and adjust flow to desired rate by monitoring the supply pressure indicators.

   g. Center the four way valve in the through position and close the supply needle valve in the hydraulic system.

   h. Start hydraulic pump.

   i. Start main drive motor and check machine over to ensure proper operation.

2. HYDRAULIC SYSTEM LOADING PROCEDURE:

   a. Check to see that suppy valve is fully closed.

   b. Place selector valve in load position.

   c. Open supply valve and monitor strain indicator. CAUTION Do not open supply valve very much until positive load indication is observed. This is to prevent overloading the system with a sudden surge of pressure.

d. Monitor pressure indicator and strain indicator.

e. Adjust supply valve until desired loading is attained.

3. UNLOADING PROCEDURE FOR HYDRAULIC SYSTEM:

a. Place selector in bypass position and close supply valve.

b. Place selector valve in unload position and open supply valve to complete unloading.

c. Monitor strain indicator; when load is decreased to desired level place selector in the bypass position.

4. SHUTDOWN PROCEDURE:

a. Unload hydraulic system.

b. Secure hydraulic pump.

c. Secure power to heaters.

d. Secure power to drive motor.

e. After the machine comes to a complete stop close the lubrication system supply valves.

f. Secure power to the pumps.

g. Shut off water supply to heat exchangers.

h. Secure all remaining electrical power.

Experience gained from operating the machine will undoubtably alter these procedures but for the initial phase of operation these instructions will provide a guide for safe running.

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13. ABSTRACT

The gear or lubricating oil test instrument described incorporates the four square principle in its design. This basic configuration is improved over like devices in that loading may be varied while the machine is in operation and rotating seals have not been used. The elimination of the need for these seals allows prolonged operation at high speeds. The machine will be capable of operation at 10,000 RPM however, the first phase of operation will call for a speed of 1800 RPM. The device has two separate and complete thermostatically controlled lubrication systems; this gives the option of testing lubricants in the main, as well as the test section. The test section is provided with an observation port which is easily removed to change or service the test gears. The rig as a whole was designed for reliability of operation and ease of maintenance.

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Four Square						
Lubricant Test Instrument						
Gear Test Instrument						

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